POWER

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Truth

IF EVERYBODY knew the truth about everything, it would be easy to know what was right and just and best for the common good.

It would be mighty disagreeable for some people, but that is usually their fault.

In those cases where it is better that the truth be not known, it is usually some individual who profits by the suppression. And if that individual interest can be served to real advantage without detriment to others, deceit may become Charity or Mercy.

Who would condemn the physician who keeps alive the spirit of hope in a patient whom he knows is beyond help?

Who does not applaud the lie deliberately, courageously, told by Sister Simplice, the nun who had never lied in her life, to the vengeful pursuer of Jean Valjean?

It is often the case, as Pope says, that "Blunt truths more mischief than nice falsehoods do."

Is it not after all a question of motive?

The man who deliberately deceives for his own advantage or to cover up an ignoble action by another is contemptible, and the proven liar is abhorred of men.

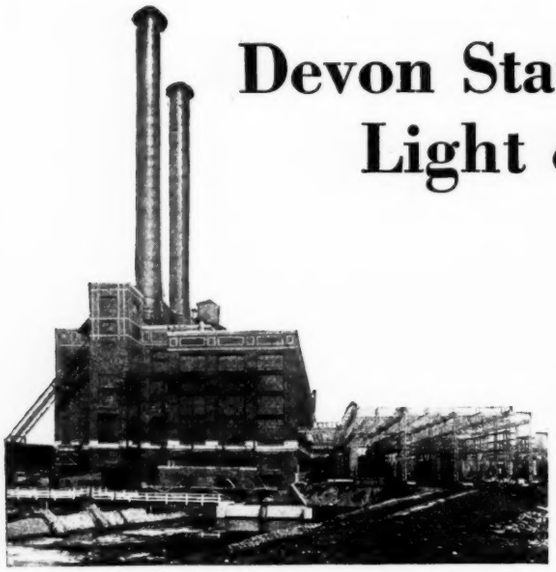
The man who tells the truth, no matter how hard it hits him, commands admiration and invites forgiveness; and a reputation for veracity is as much fine gold.

Sophistry—truth purposely told in such a way as to be misunderstood or, being known to be misunderstood, allowed to go uncorrected—is as reprehensible as deliberate falsehood.

Disagreeable truths may well be let alone unless some useful purpose can be served by their exposure. To deny them when exposed demands stronger grounds for its justification.

If you are perplexed, hold fast to the Truth, unless the motive of your departure from it is so clean and harmless and commendable that you would be proud to have it known of all men.

Devon Station of the Connecticut Light & Power Company



PRESENT installation consists of three 25,000-kva. units and six 16,800-sq.ft. boilers, but the layout provides for a possible ultimate capacity of 200,000-kva. and 16 boilers. Boilers are stokered fired, and a centralized electric system of boiler control is provided. Turbines are arranged for two stages of bleeder heating, and evaporators are used to supply makeup boiler feed from the brackish tide water.

A LARGE portion of the western half of Connecticut, a section that is highly developed industrially, is supplied, either directly or indirectly, by the Connecticut Light & Power Co. This territory is supplied through a network of 66,000-volt transmission lines interconnecting the company's new Devon station and the existing stations at Waterbury, Stevenson and Bulls Bridge, the latter two being hydroelectric plants having a capacity of 25,950 kw. (see Fig. 1). The company has expanded rapidly in the last few years to meet the increasing demands of the territory and by the end of 1924 will have an aggregate generating capacity of over 120,000 kw. Of this, 67,500 kw. is represented by the company's new Devon station, the first 22,500-kw. unit of which has recently been placed in operation.

Although the idea of building a large power station at Devon has been under consideration for a number of years, the plans were not developed until early in 1922 and ground was broken in August of that year. The site upon which the plant is being built was selected after an exhaustive study covering several years and is on the Housatonic River, near the point where the river flows into Long Island Sound. The tract consists of about sixty acres adjoining the main line of the New York, New Haven & Hartford Railroad, so that both rail and tidewater coal can be received. The chief factors influencing the selection of the site were ample condensing water, coal supply and storage, reasonable proximity to the load center and facility of obtaining suitable location for the several transmission lines that will radiate from the station.

From a glance at the map of Connecticut there would appear to be many places either on Long Island Sound or on the various rivers of the state where suitable sites could be found for large steam plants. The exhaustive study which the company made of all of these locations indicated there were only three places in the state where adequate condensing water and other suitable conditions could be found for a plant of this character —on the Thames River in the eastern part of the state, on the Connecticut River in the central part and on the Housatonic River in the western section. With the completion of Devon there will be a large station located at each of these three points, which are about 50 miles distant from one another.

In the design of the station the engineers were guided by a number of factors. First of all, the urgency of additional power to take care of the company's load required that the plant have as high a degree of reliability as modern standard practice could give. Part of the time the station will be used on base-load service and at other times to supply peak loads, depending upon the amount of water power available. For at least six months in the year, January to June, there is sufficient water to carry all the base load on the system and part of the peaks. This operation precluded the economic use of some refinements that are being used in some of the base-load steam stations. It was desirable to keep the investment cost as low as possible consistent with reliable design, at the same time keeping in mind that a change in fuel costs and load conditions might make desirable changes to obtain a higher efficiency than is possible with the present equipment. Furthermore, a number of the present refinements in power-plant design cannot be considered out of the experimental stage, and future experience might also make them economically desirable in this plant.

For the present, provisions have been made to receive

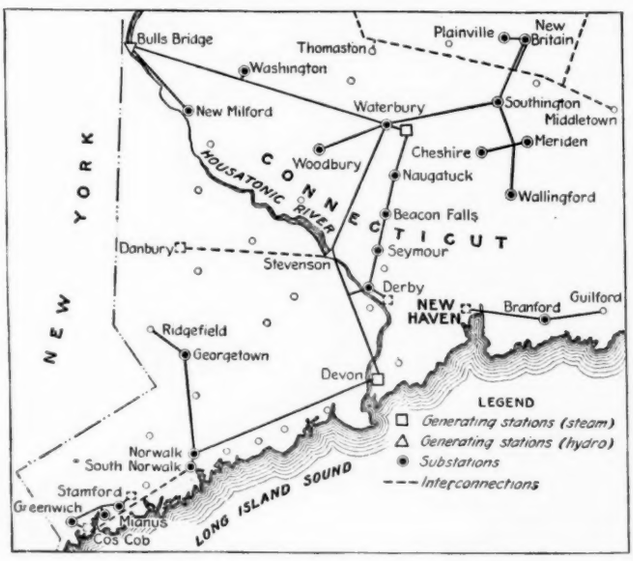


Fig. 1—New plant is located at Devon

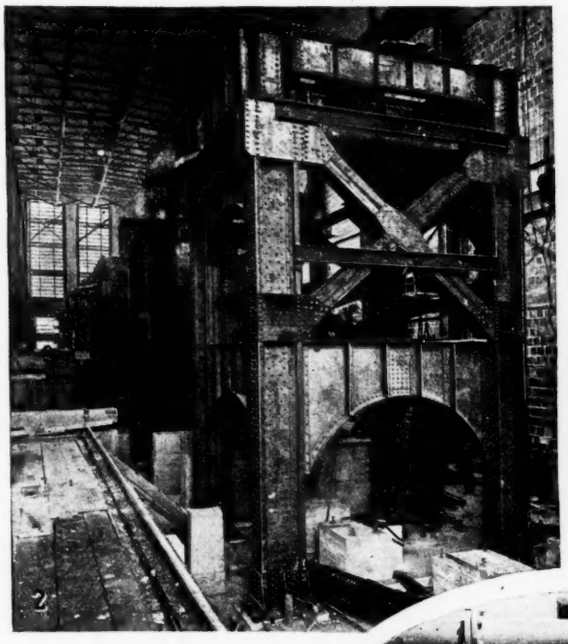


Fig. 2—Steelwork in turbine foundations

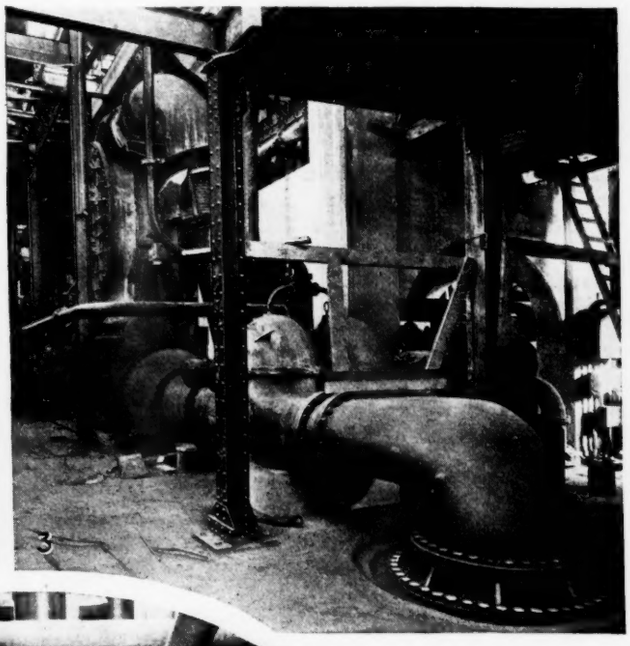


Fig. 3 — One 40,000-sq.ft. condenser on each 25,000-kva. turbine

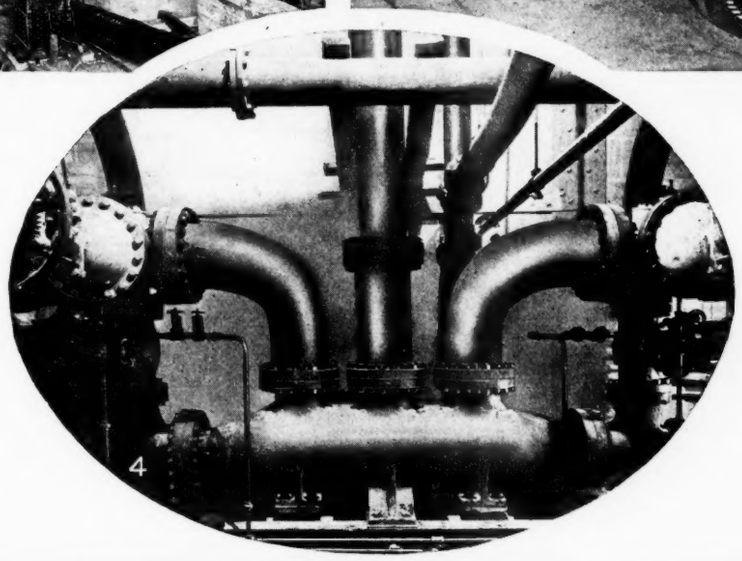
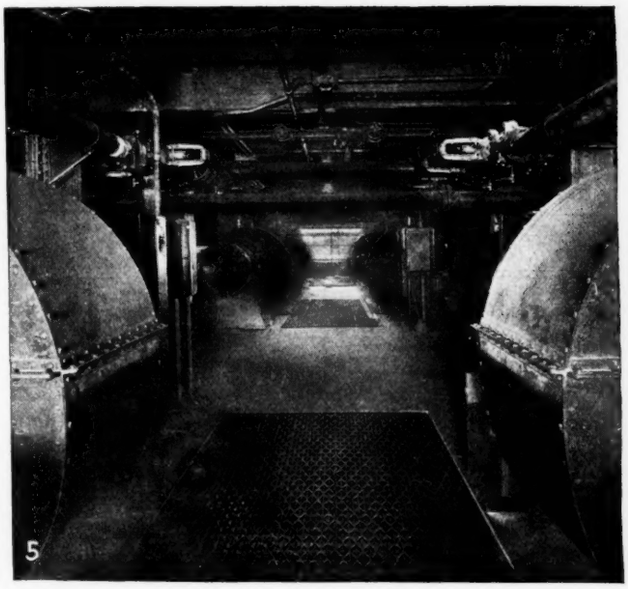


Fig. 4 — Manifold in main steam piping between boilers and turbines

Fig. 5—Forced-draft fanroom below boiler firing aisle



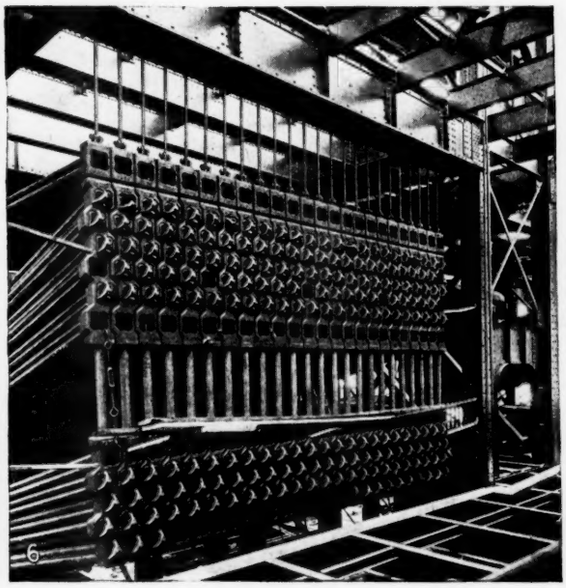
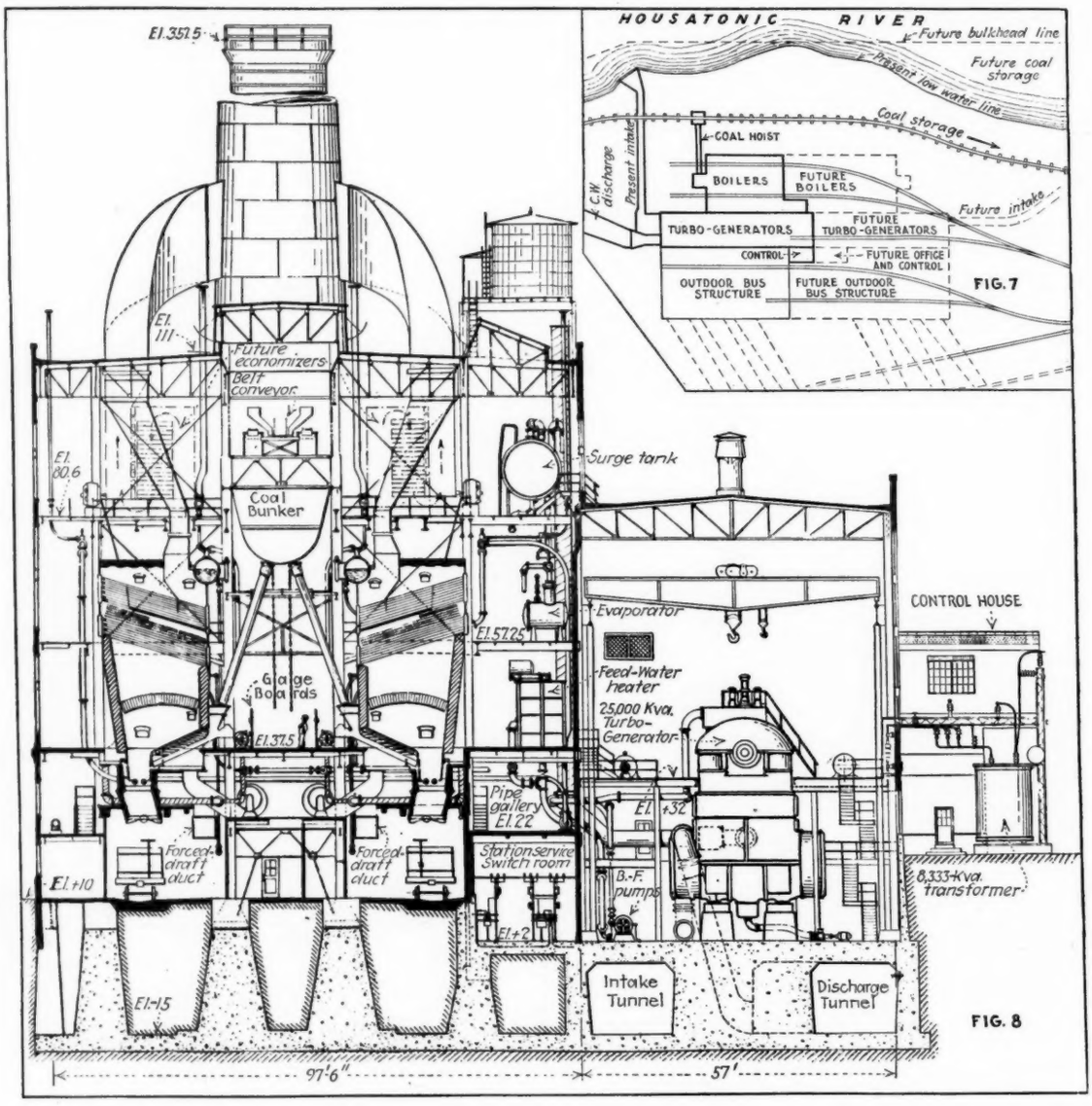


Fig. 6—One of the 16,800-sq.ft. boilers under construction; superheater located between tenth and eleventh rows of tubes

coal by rail only. The coal trestle, which is maintained at the elevation of the main-line railroad tracks, is carried through the power-station yard. It is estimated that under this trestle 40,000 tons of coal can be stored, which, with 10,000 tons storage alongside the trestle, is the present storage requirements. Provision has been made in the design of this trestle for the location of a track scale.

The only equipment provided at present for reclaiming coal is a locomotive crane. It is expected that some time in the future a dock will be built along the Housatonic River approximately parallel to the present coal trestle. The design for the future unloading has not been worked out as yet, but it is contemplated when the station grows to a sufficient size to warrant the expense, a bridge crane will be utilized. The ultimate development will provide storage for over 100,000 tons of coal.

A track hopper is directly beneath the trestle and over the two coal crushers, so that coal passes from the cars into the hopper and to the crushers. Two skip hoists, which receive the coal from the crushers, have their lower ends immediately below the receiving hopper on the crushers, and have a capacity of 150 tons per hour. These skips are of the double-balance type, each provided with a separate hoisting machine. Two belt conveyors, into which these hoists dump after passing through a receiving hopper and automatic loaders, also have a capacity of 150 tons per hour. The belt conveyors run the entire length of the boiler room above an overhead bunker. Each belt conveyor is equipped with a weightometer.



Figs. 7 and 8—Property plan and elevation through plant

In the present plant there are six boilers installed. Four boilers are considered the full complement per generating unit. In the ultimate plant, however, there will be four groups of four boilers, with six turbines, it being assumed that this total number of boilers will have sufficient capacity to handle five generating units. The sixth machine is to be considered a spare unit. The boilers are arranged on two sides of a single firing aisle. Four boilers, two on each side of the firing aisle, connect to a common stack, the top of which extends up

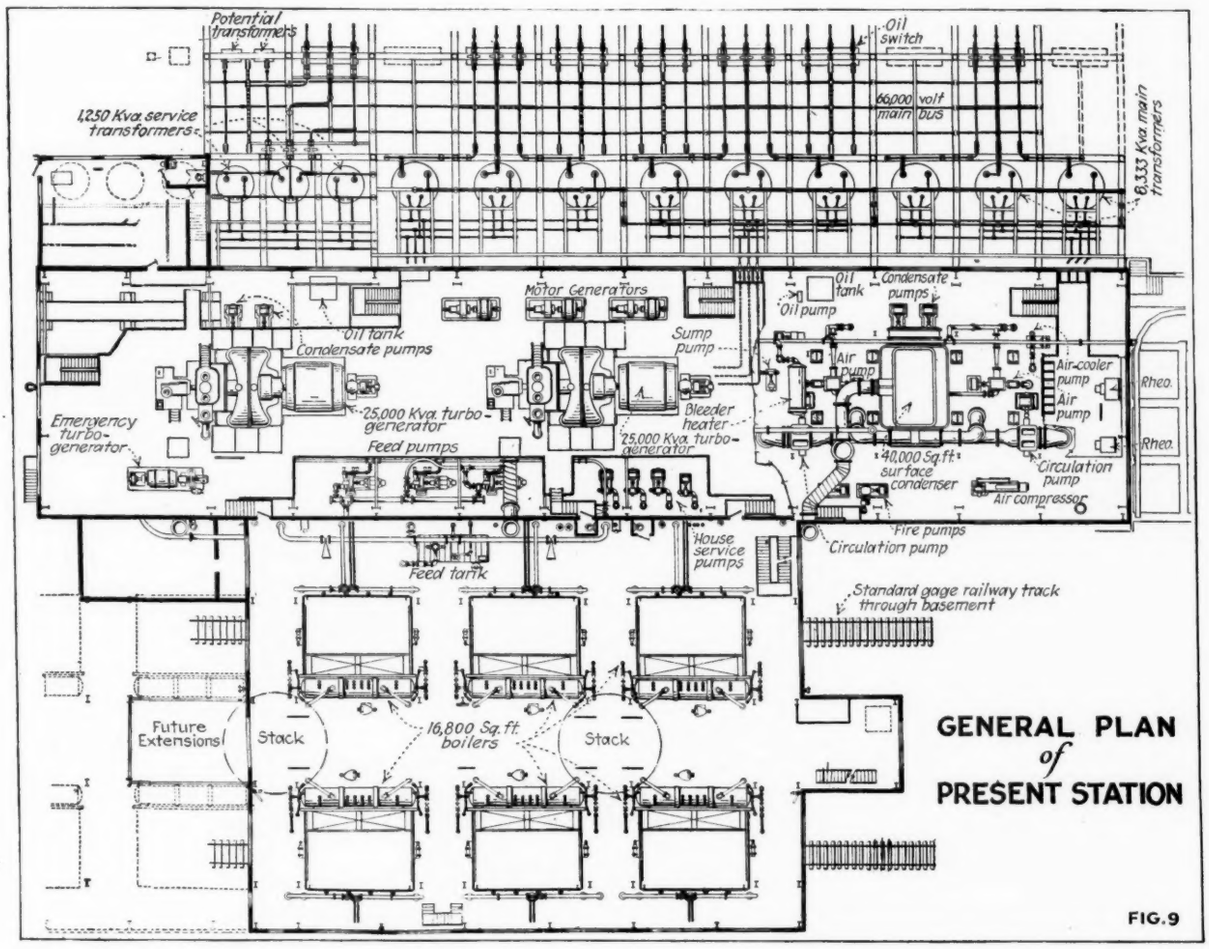
to 320 ft. above the firing-aisle floor and 347 ft. 6 in. above the yard level. Each boiler is equipped with a 12-retort 27 tuyere underfeed stoker, designed to burn a maximum capacity of 1,400 lb. of coal per retort per hour. The stokers are equipped with double-roll clinker grinders and agitators are installed in the stoker magazines. Coal is delivered from the overhead bunker direct to the stoker magazines through swinging-down spouts. In addition to the weightometer on the belt conveyors, every group of three retorts of the stokers has installed revolution counters. Individual direct-current forced-draft fans are provided. In order to provide the equivalent of a spare fan, an emergency air duct is installed under each row of boilers connecting together the fans located on the corresponding side of the center line of the boiler room.

The boilers are of the horizontal water-tube cross-drum type, having 16,800 sq.ft. of heating surface each and designed for 320 lb. gage working pressure. They are 22 tubes high by 48 tubes wide, with an inter-deck superheater, of 1,960 sq.ft. heating surface, between the tenth and eleventh rows of tubes, designed to give a normal superheat of 200 deg. Every other tube in the bottom four rows has been omitted, in order to form a slag screen. The building is laid out so that at a later date economizers can be installed above the boiler. For this reason the present uptake breechings are of a temporary character.

The main steam piping is laid out so that each group of four boilers discharge into a common manifold, no steam header being used (see Fig. 10). On the first manifold two outlets go to the first two turbines and the third connects to the crossover line going to the adjacent manifold. On the second manifold one outlet goes to the third turbine, one connects to the crossover line from the first manifold and the third will later connect to the crossover line to the third manifold. These manifolds are the anchor points of the main steam piping. Vanstone joints are used throughout.

A centralized electrical system of boiler-combustion control is installed. In this system a master regulator is installed at some point on the steam lines and fluctuating steam pressure causes this regulator to act upon a motor-generator set. This motor-generator set consists of a single motor and two generators, the first generator supplying the field current for the direct-current driven forced-draft fans, and the second generator supplying the field current for the stoker motors. As the load varies on the boilers, the speed of the stokers and forced-draft fans is changed to meet the new conditions. In this way parallel operation of the boilers is obtained and the ratio of fuel and air is automatically maintained throughout the range of load. In addition to controlling the speed of the stoker and fan motors a damper controller is used, which governs the opening of the rear damper on the individual boilers by pressure over the fire.

There are two sources available for feed-water supply, one being purchased water from the local water company and the other the brackish river water. In



either case it would be necessary to provide some means of keeping the solids in the water out of the boilers, therefore it was decided to install evaporators and use the river water, as this proved to be the most economical. Two high-pressure two-effect evaporators have been installed of sufficient capacity so that one can take care of the normal demands of the station and the other is used as a spare.

On account of the tide in the river the percentage of solids in the water at the plant varies over wide ranges. On one test of the water, 1,705 grains of solids per gallon were found at high tide and 55 grains at low tide. To obtain water for the evaporators with as low a percentage of solids as possible, water is pumped from the river into two 15,000-gal. tanks on the boiler-house roof during low tide, and used during the period of high tide. In this way the blowdown from the evaporators is kept at a minimum. In addition to the two storage tanks for river water, city water is stored in two 4,000-gal. tanks as an emergency feed-water supply. For the general station service water is pumped from the river into a 15,000-gal. tank.

All auxiliaries are electrically driven with the exception of the three boiler-feed pumps. The general heat balance scheme is to pump the condensate from each main turbine through its two stages of bleeder heating and then to the open feed-water heater where the heat in the exhaust from the boiler-feed pumps is absorbed. A future article will describe the heat balance and boiler settings.

Three turbines of the single-flow Bauman design, rated at 25,000 kw. make up the initial installation. These are equipped for two-stage bleeder heaters, the bleeder points being taken off at approximately 3.2 lb. and 13 lb. absolute when the machines are operating at 16,000 kw. which is the point of highest efficiency.

In the selection of the turbines the engineers had the choice of units of two different water rates. One of these had a somewhat higher efficiency than the other, but with a water-rate curve that sloped off more rapidly from the point of best efficiency than the machine with the lower efficiency. Owing to the wide range of load that the machines will have to operate under at different times during the year, on account of being connected in parallel with water-power stations, machines with the slightly lower but flatter efficiency curve were selected. These machines have a water rate of 11.5 lb. at 10,000, 10.7 at 16,000, 10.85 at 20,000 kw., with a steam condition at the throttle of 285 lb. gage and 200 deg. superheat, expanded down to 28.5-in. vacuum.

Each turbine exhausts into a 40,000-sq.ft. surface condenser equipped with 1-in. tubes, and is of the radial-flow type with divided water boxes. Two circulating pumps serve each condenser; one pump operating alone is of sufficient size (28,000 gal. per min.) to give a 28.5-in. vacuum with 60-deg. water, or 28-in. vacuum with 70-deg. water. The two pumps operating together (43,000 gal. per min.) will maintain a 28.5-in. vacuum with 70-deg. water, or 28-in. vacuum with 80-deg. circulating water. There are also two condensate and two air pumps provided on each main unit. One of the condensate pumps has sufficient capacity to handle the condensate under maximum conditions. The air pumps are designed for 10 cu.ft. of free air at 70-deg. F. and 28.5-in. vacuum in the condenser.

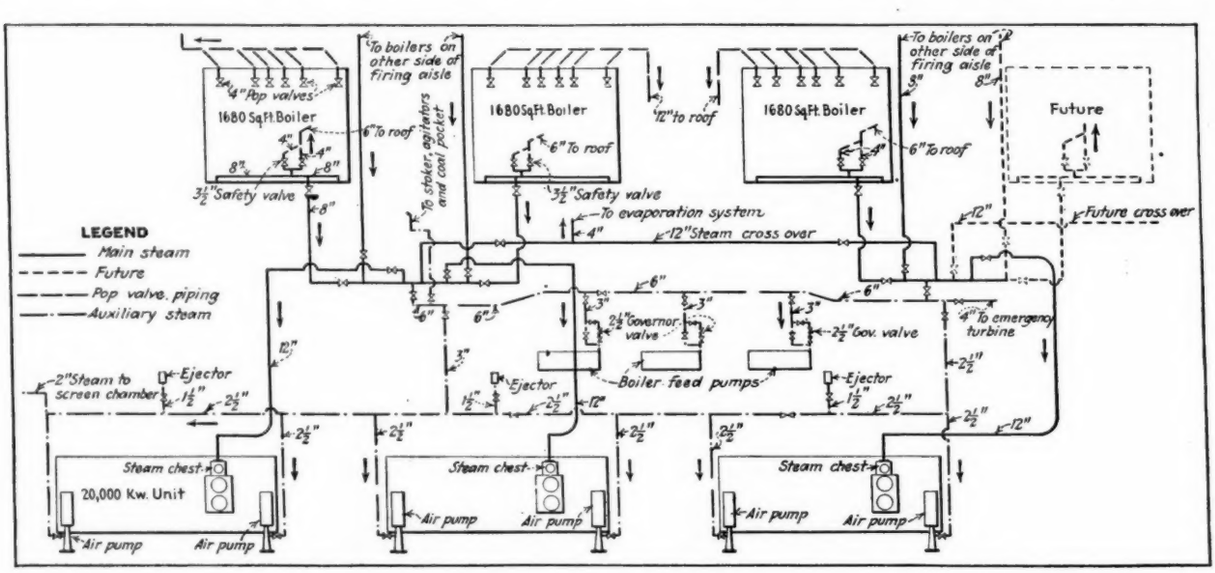


Fig. 10—Layout of main high-pressure piping system

The present generator installation consists of three 25,000-kva. 13,900-volt 60-cycle 90 per cent power factor machines, or an aggregate generating capacity of 75,000 kva. The layout provides for a possible ultimate capacity of 200,000 kva., in which event the station will be extended in the future with three 40,000-kva. machines.

Electrically, the station has many features of great interest. The keynote of the electrical layout is simplicity in design and operation. The principal feature is the elimination of a bus at the generating voltage of 13,900 volts, the generator leads connecting directly to the low-tension terminals of the step-up transformer banks in the outdoor bus structure, where the current is stepped up to 66,000 volts. This practically means that 66,000 volts becomes the generating bus voltage. Each unit has differential-relay protection on the generator and the transformers. Internal trouble on either the generator or the transformers will trip the circuit breaker on the high-tension side of the transformer

bank, the overspeed device on the turbine and the generator field breaker.

The general contractors for the Devon station were the J. A. P. Crisfield Contracting Co., a subsidiary of the U. G. I. Contracting Co., Philadelphia. The design and construction were carried out under the engineering supervision of Paul Spencer, consulting engineer, and K. M. Irwin, his mechanical engineering assistant, Irvin W. Day, vice-president, and E. J. Amberg, electrical engineer of the Connecticut Light & Power Co. C. F. Hirshfeld, Detroit, Mich., acted as advisory consulting engineer for the latter organization.

DATA ON PRINCIPAL EQUIPMENT IN DEVON STATION OF CONNECTICUT LIGHT & POWER CO.

GENERAL

Location of plant	Devon, Conn.
Character of service	Light and power
Capacity, present building, kva	75,000
Ultimate, kva	200,000
Boiler house, ft	98x130
Turbine room, ft	57x255
Control house, ft	27x40
Width of firing aisle, ft	20
General contractors	The J. A. P. Crisfield Contracting Co.

BOILERS AND SUPERHEATERS

Manufacturers	Springfield Boiler Co.
Type of boiler	Sectional, water-tube, cross-drum
Number installed	6
Steam-making surface per boiler, sq.ft.	16,800
Ratio steam-making surface to projected grate area	54 to 1
Furnace volume, cu.ft	5,700
Sq.ft. of steam-making surface per cu.ft. of furnace volume	2.95
Cu.ft. of furnace volume per sq.ft. of projected grate area	18.3
Arrangement of tubes	22 tubes high and 48 wide
Sizes of tubes	3 in. x 20 ft. No. 10 gage
Boiler pressure, lb. gage	320
Type of superheater	Elesco
Manufacturer	Superheater Co.
Location of superheater	Between 10th and 11th rows of tubes
Superheating surface per boiler, sq.ft	1,960
Ratio superheating to steam-making surface	1 to 8.57
Steam temperature at 200 per cent rating deg. F	628
Corresponding superheat, deg. F	200

FUEL

Kind	Pennsylvania bituminous
Heat value, B.t.u. per lb	13,500

STOKERS

Manufacturer	Westinghouse Elec. & Mfg. Co.
Type	Underfeed
Number per boiler	1
Width	21 ft. 3½ in.
Actual grate area, sq.ft	333.27
Projected grate area, sq.ft	311.27
No. of retorts per stoker	12
Pounds of coal burned per sq.ft. of grate area at 200 per cent of rating	32.3
Coal agitators	Negus Tiffany

Efficiencies at different boiler ratings:

Per Cent of Rating	Combined Boiler and Furnace Efficiency, Per Cent
100	76
150	79
200	78
300	70

BOILER-CONTROL SYSTEM

Forced-draft fans and motors, stoker motors, damper regulators, automatic combustion control system, boiler panels including boiler meters, draft gages and tachometers, supplied by The Benjamin Engineering Co.

Forced-draft fans, manufacturer	B. F. Sturtevant Co.
Number of fans	6
Capacity per fan, cu.ft. per min	65,000 at 7 in. static pressure
Stoker and fan drives	General Electric d-c motors
Boiler meters, draft gages and tachometers	Bailey Meter Co.

STACKS

Manufacturer	The Porcupine Co.
Number	2
Type	Self-supporting steel, lined for full length
Height	320 ft. above firing-aisle floor
Diameter inside	25 ft. at base, 18 ft. at top
Area at top, sq.ft	254
Boilers per stack	4
Ratio of stack area (at top of stack) to steam-making surface	1 to 264
Ratio of stack area (at top of stack) to projected grate area	1 to 4.85
Stack lining, hard burned hollow radial brick	M. W. Kellogg Co.

BOILER-FEED PUMP

Manufacturer	Ingersoll-Rand Co.
Number of pumps	3
Type	Cameron centrifugal, 3-stage, 350-lb. discharge
Capacity, e	750 gal. per min.
Drive, steam	Westinghouse Elec. & Mfg. Co.

EVAPORATORS

Manufacturer	Griscom-Russell Co.
Number	2
Type	High-pressure, 2-effect
Capacity, each	9,000 lb. per hr.

HEATERS

Manufacturer	H. S. B. W.-Cochrane Corp.
Number and type	One, open square with Cochrane Weir and Simplex measuring device
Capacity	600,000 lb. per hour

COAL HANDLING

Type	Belt conveyors and skip hoist
Furnished by	Link Belt Co.
Capacity	120 tons per hr.
Crushers, number	2
Crushers, make	Link Belt Co.
Overhead bunker, capacity and type	700 ton, steel
Bunker furnished by	The Porcupine Co.
Coal distributed in bunker by	2 belt conveyors each equipped with tripper
Capacity, each	60 tons per hr.
Yard, coal storage	Present, 50,000 tons, future, 100,000 tons
Method of reclaiming from storage	Locomotive cranes (temporary only)
One locomotive crane	McMyler Interstate Co.
One steam shovel No. 29 caterpillar tread	Osgood Co.

ASH HANDLING

Make	Baker Dunbar Co.
Ash storage per boiler hopper, cu.yd	17
Gates (air operated) per hopper, number and size	Two 5 ft. 6 in. x 6 ft. 6 in.

MAIN GENERATING UNITS

Manufacturer	Westinghouse Elec. & Mfg. Co.
Type of turbines	Bauman single-flow
Capacity, each	22,500 kw. at 90 per cent power factor
Number of units	3 (present)
Speed, r.p.m	1,800
Throttle pressure and superheat	285 lb. gage and 200 deg. F.
Type of generator	3-phase, 60-cycle, 13,900-volt
Ventilation	Closed system with Griscom Russel Co. U-fin air cooler

Guaranteed performance of turbines: Steam condition 285 lb. gage, 200 deg. F. superheat, expanded to 28.5 in. vacuum

Load, kw.	Lb. of Steam per Kw.-Hr.
5,000	13.83
10,000	11.5
16,000	10.7
25,000	11.25

EMERGENCY TURBINE

Manufacturer	Westinghouse Elec. & Mfg. Co.
Number	One
Size	1,000-kw. 3-phase, 60-cycle, 2,300 volt
Use	Emergency auxiliary power

SURFACE CONDENSERS

Manufacturer	Westinghouse Elec. & Mfg. Co.
Number	3
Tube surface, each, sq.ft	40,000
Sq.ft. of tube surface per kw. of generator rating	1.8
Connection to turbine	Rigid
Support	Springs
Condenser tubes	American Brass Co.
Condenser tubes	Chase Metal Works
Condenser tubes	Scovill Mfg. Co.

CIRCULATING WATER PUMPS

Manufacturer	Westinghouse Elec. & Mfg. Co.
Number per condenser	2
Capacity, gal. per min	28,000 against a 25-ft. head
Drive	2,300-volt a.c. motors

AIR-PUMP

Manufacturer	Westinghouse Elec. & Mfg. Co.
Number per condenser	2
Type	Le Blanc
Capacity cu.ft. per min	10
Drive	2,300 volt a.c. motor

CONDENSATE PUMPS

Manufacturer	Westinghouse Elec. & Mfg. Co.
Number per condenser	2
Type	Centrifugal
Drive	2,300 volt a.c. motor

DATA ON PRINCIPAL EQUIPMENT IN DEVON STATION [CONCLUDED

SCREEN WELL AND INTAKE TUNNELS

Screen wells, number	3
Three traveling screens	Chain Belt Co.
Width each screen	10 ft. 2 in.
Travel	10 ft. per minute

MISCELLANEOUS EQUIPMENT AND MATERIALS

Equipment	Manufacturer
Building steel	McClintic Marshall Co.
Pintex hollow tile	Wilkes-Barre Clay Production Co.
Steel sash and skylights	Detroit Steel Products Co.
Gratings for walkways	Irving Iron Works
Firebrick	Harbison Walker Refractories Co.
Bernitz perforated furnace blocks	Bernitz Furnace Appliance Co.
Carbofrax brick	Carborundum Co.
Liptak tiles	Liptak Fire Brick Arch Co.
Sil-O-Cel brick	Celite Products Co.
High-temperature cement	Quigley Furnace Specialty Co.
High-test cement	Hy-Test Cement Co.
Station crane	Northern Engineering Works
Side-dump ash cars	Western Wheeled Scraper Co.
Soot blowers	Vulcan Soot Cleaner Co.
Copes feed-water regulators	Northern Equipment Co.
Stets feed-water regulators	Williams Gage Co.
Air compressors	Bury Compressor Co.
Oil-filtering and storage system	S. F. Bowser & Co.
Soot conveyor	American Conveyors Corp.

Steam, water and air valves: Consolidated Safety Valve Co.; Yarnall-Waring Co.; Lewis Steam Specialty and Valve Co.; Crosby Steam Gage and Valve Co.; Crane Co.; Nelson Valve Co., and Schutte & Koerting Co.

Piping and fittings: Weatherly Foundry & Mfg. Co.; H. M. Treadwell Co.; Whittney McDonal Co.; Crane Co.; M. J. Daugherty Co.; Reading Steel Foundry Co.; Kutztown Foundry & Machine Co.; General Fire Extinguisher Co.; and American Spiral Pipe Co.

Expansion Joints: Ford Brothers; Jos. Oat, and U. S. Rubber Co.

Recording and Indicating Gages and Thermometers: Crosby Steam Gage & Valve Co.; The Bristol Co.; Leeds & Northrup Co.; Schaeffer & Budenberg Co.; Bailey Meter Co., and Schade Valve Co.

Steam-flow meters: General Electric Co.

Packing: Garlock Packing Co.

Load indicating, signaling system and control for motor operated valves: Payne-Dean Ltd.

Silent chains: Morse Chain Co., and Ramsey Chain Co.

Breeching and air ducts: The Porcupine Co.

Pipe covering (85% magnesia and Hi-Temp.) Philip Carey Co.

Strainers: Elliott Co.

Surge and storage tanks: New York Eng. Co.; G. Woolford Wood Tank Co., and E. F. Schlicter

Heating equipment: H. B. Smith Co.; Nash Eng. Co., and Grinnell Co., Inc.

Other important electrical equipment supplied by Westinghouse Elec. & Mfg Co.; General Electric Co.; Western Electric & Mfg. Co.; Condit Electric Co.; Electric Power Equipment Corp.; Leeds & Northrup; Roller Smith; Cutler-Hammer Mfg. Co.; Niagara Improvement Co.; Jeffery DeWitt Insulator Co.; Locke Insulator Co.; Electric Storage Battery Co.; American Brass Co.; R. D. Coombs Co.; Erie Fittings Co.; V. V. Fittings Co.; Kerite Insulated Wire & Cable Co.; Bishop Gutta Percha Co.; Standard Underground Cable Co.; National Conduit & Cable Co.; John A. Roebling's Sons Co.; Johns-Manville, Inc. Wheeler Mfg. Co.; and States Co.

Operation and Care of Cam-Type Manual Controllers for Electric Motors

By H. D. JAMES

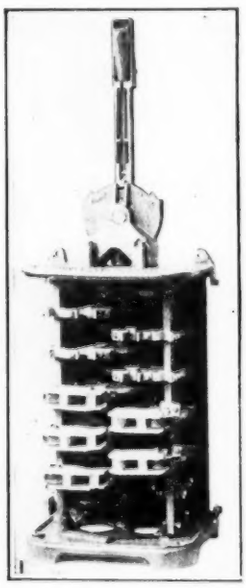
Manager, Control Engineering Department, Westinghouse Electric & Manufacturing Company

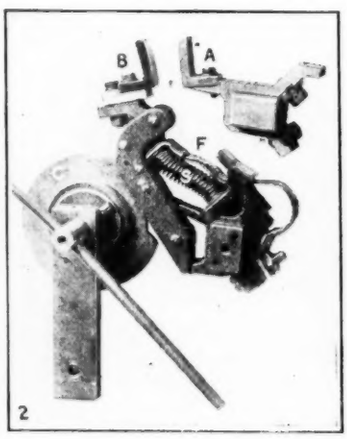
CAM-TYPE controllers have the same general appearance as drum controllers and are mounted and operated in the same manner, but the contact element is different. In place of the rotating cylinder of the latter, there is a camshaft which engages a series of contactors which replace the drum fingers of the drum-type controller. The contact mechanism is illustrated in detail in Fig. 2. It consists of a stationary element *A*, which may or may not be provided with a blowout coil, and a movable element *B*, held open by a spring *S* and closed by a cam *C*. The action of this contact is exactly the same as that of the magnetic contactor, and the same contacts are used.

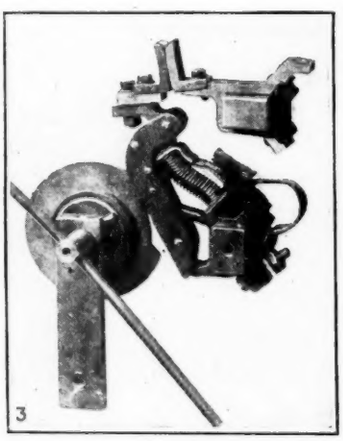
Figs. 2 to 4 show the three positions of the contact: First, when open; second, when just touching at the tip; third, when it is entirely closed, making contact at the heel. In opening, the process is reversed. This rolling action from the tip to the heel and back again in closing and opening causes the arc to be broken at the tip and the current to be carried at the heel. By using rolling action instead of a sliding action in closing, the mechanical wear is eliminated and much heavier pressures can be used.[1]

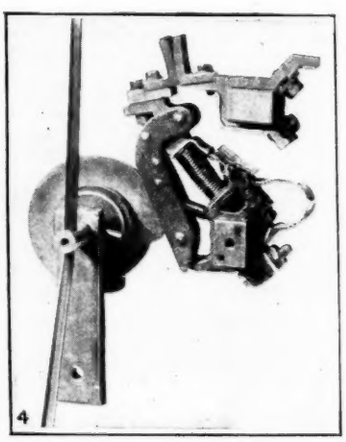
The current is carried from the movable contact to the terminal by means of flexible shunts; one end of the spring is insulated so that no current can pass through this member. Because the spring is a considerable distance from the contact, it is not directly

[1] A more complete discussion of this contact is given in *Power*, May 31, 1921, page 878.









Figs. 1 to 4—Cam-type manual controller and contacts in open and closed position

Fig. 1—Cam-type manual controller. Fig. 2—Contactor in open position. Fig. 3—Contactor partly closed. Fig. 4—Contactor completely closed. The last three illustrations show the action of the contactor when closing. When closing, the contactor first engages at the tips, or outer portion, and rolls down to the heel. In opening, the action is reversed.

affected by the temperature of the contact or the arcing, and is, therefore, in no danger of having its temper drawn owing to excessive heat.

Various combinations of switches can be obtained by changing the number of contactors and the shape of the cams. The length of the frame can be adapted to the desired combination by changing the length of the insulating bars and the sheet-iron cover.

The following points should be observed when inspecting a cam controller:

1. The contacts, when closing, are engaged at the tip or upper edge and roll down to the heel or bottom of the contact. The reverse action should take place when opening. Examine the contacts to see that this action takes place, and when the contacts are in their final closing position, see that they engage throughout their entire width.

2. Do not file the contact surfaces, as the wear due to operation improves the contact surfaces by eliminating the minor high spots. Ordinarily, a contact will carry more current after it has been "worn in" than when new. If the action of the contact is correct, the burning should all take place at the tip.

3. When the contacts have burned back so that the arcing occurs close to the heel, both contacts should be renewed. Do not attempt to use an old contact in combination with a new one.

4. Do not lubricate the contact surfaces. They are so designed that very little rubbing action takes place, which makes lubrication unnecessary and it is a disadvantage on account of the dirt that tends to accumulate on lubricated surfaces.

The operation of roller pin, hinge pin and camshaft bearings will be improved by a small amount of lubrication unless the air is very dirty. Lubrication tends to hold dirt in the bearings, and under some conditions they will operate better if dry.

5. Measure the contact pressure with a spring balance in a manner similar to that described in the article, "Operation and Care of Drum-Type Manual Controllers," Mar. 4. Each manufacturer should furnish a table showing the normal contact pressure. This should be maintained in order to secure the best operating conditions. A low pressure will increase the heating, and a high pressure will make the controller hard to operate.

6. The controller notches are indicated by a ratchet wheel. In many controllers the pressure of the roller in the ratchet wheel can be adjusted. The operation of the controller can often be materially improved by the careful adjustment of the ratchet spring. Where the air is free from grit, the ratchet wheel and roller should be lubricated. Examine all the bolts to see that they are tight. This is particularly necessary on bolts that hold the contacts, the shunts and terminals in place. A loosening of any of these bolts may materially increase the heating of the controller.

7. If the controller is exposed to corrosive fumes which discolor the copper parts, it will be necessary to clean the contacts with fine emery cloth or sandpaper, and it may be necessary to disconnect the current-carrying joints and clean them from time to time. Sometimes the covering of these joints with a thin coating of solder will prevent corrosion.

8. The controller should be cleaned out from time to time, depending upon the rapidity with which dirt accumulates within the controller. A convenient method is to use compressed air at low pressure. Do not subject the controller parts, particularly the insulation, to a high-pressure air jet, as this may injure the insulation.

Both drum-type and cam-type controllers can be provided with either a rotating handle or a lever handle. They may also be mounted vertically on the rear of a panel and operated by a hand-wheel from the front through suitable gearing. If the controller is mounted back of the panel with the shaft horizontal, it can be extended directly through the panel without the use of gearing.

Cam controllers are frequently built motor-operated. The construction of the controller is the same as for manual operation, the handle being replaced by a reduction gear and pilot motor.

The drum controller is more compact for certain switching combinations, such as reversing. On the other hand, the cam is more easily designed for complicated switching, and new combinations can be more easily made as these combinations are accomplished by using cams of different shapes, while in the drum design new drum castings are usually required. The cam units can be removed individually without dismantling the controller. The same is true of the contact fingers of the drum controllers. It is usually necessary to take the drum out of the controller in order to replace any of the drum segments, and after replacement, these segments should be trued up in a lathe.

Why the D Slide Valve Has Lap and Lead

One of the first problems encountered by the beginner in steam engineering is that of the action of the slide valve. If one will go through the cycle of events covered by the valve, its action is easily understood.

The simplest form of slide valve is shown in Fig. 1 and consists essentially of a hollowed block of cast

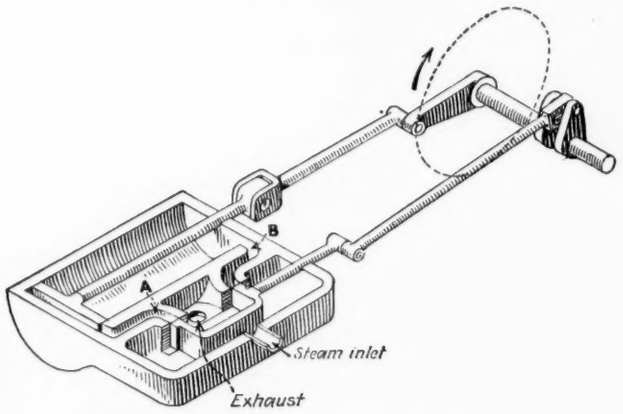


Fig. 1—Elementary slide-valve engine

iron. It is set over the steam ports *A* and *B* and is moved by a small crank, which is set 90 deg. ahead of the engine crank. With the piston at the left-hand end of the cylinder, Fig. 1, the engine crank connected to the piston being at dead center, the valve crank is vertical. Under these conditions the valve is at the middle of its stroke and covers both steam ports. The slightest displacement of the piston will rotate the shaft in the direction shown, and the motion of the valve

crank will shift the valve to the right, admitting steam down port *A*. The piston is forced to the right, turning its crankshaft, which in turn moves the valve crank to the right until the valve crank reaches dead center, as indicated in Fig. 2*A*. The engine crank is now at midposition with the piston as shown. Further rotation of

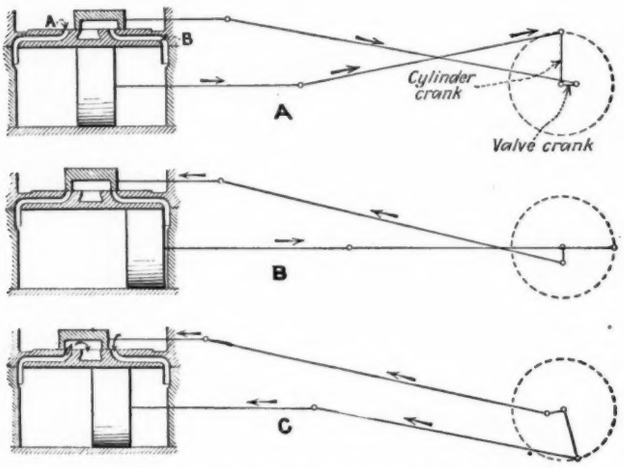


Fig. 2—Action of valve without lap

the shaft causes the valve crank to move on past dead center, and this moves the valve to the left, while steam pouring in through port *A* forces the piston toward the right. This continues until the valve covers the steam port, which takes place when the valve crank reaches midposition, Fig. 2*B*, which shows that at this time the piston has reached the end of the stroke. The valve, if shifted slightly to the left, uncovers the crank-end steam port *B* and allows steam to enter the crank end of the cylinder; this causes the piston to move to the left. At the same time the exhaust port is put into communication with the head end, allowing the steam in the head end to pass out into the exhaust line.

The plan of feeding steam to the cylinder during the entire stroke is uneconomical, as no use is made of the expansive power of the steam. While this method is of necessity in a steam pump where there is no flywheel to even up the load, it is never found in a steam engine.

If we add the lip *a* to the valve, the valve will not start to uncover the steam port *A* when its crank is at midposition and the piston at dead center. The valve, as indicated in Fig. 3*A*, must move a distance beyond the midposition equal to the added lip *a*, and the valve crank will be in the position shown, and the piston will have started on its stroke, the crank being past dead center. The valve is at the end of its stroke at Fig. 3*B* and is ready to move back, cover port *A* and uncover the crank-end steam port *B*. In Fig. 3*C* are shown the positions of the valve crank, the piston and the engine crank when the valve re-covers the steam port *A*. The piston has not reached the end of its stroke, and the expansive power of the steam is exerted during the rest of the stroke. At the dead-center position of the engine crank and piston the valve crank is in midposition, as in Fig. 3*D*, whereupon the exhaust passage is uncovered, causing release. We have, by adding a lip to the outer edges of valve (the lip, incidentally, is called the steam lap), enabled the supply of steam to port *A* to be stopped, or cutoff to take place before dead center and so obtain some expansion of the steam. But a study of Fig. 3*A* and 3*D* reveals that the addition of the steam lap has delayed the admission of the steam until after the piston has started on its stroke. This would mean that the flywheel would be compelled to pull the piston during the early part of the stroke.

It is advisable to have the steam admitted a trifle before the piston reaches dead center, so the valve crank is shifted a little farther ahead of the engine crank. In this way the steam is admitted a trifle before dead center and cutoff is a little earlier than before. This means that the valve events take place earlier; that is, before the piston has reached the position it would have in case the valve crank was set just 90 deg. ahead of the engine crank. The amount the steam port is uncovered when the piston is on dead center is called the lead. The angle the valve crank is ahead of the engine crank beyond 90 deg. is called the angle of advance. The exhaust will take place a little before the piston reaches the end of its stroke since release occurs when the valve crank is in midposition, which means that the engine crank has not reached dead center, as outlined in Fig. 3*E*. Far from being a detriment this early release is of advantage as the cylinder is freed of steam earlier, reducing the pressure in the head end on the next or return stroke. The valve re-covers the port before the piston reaches

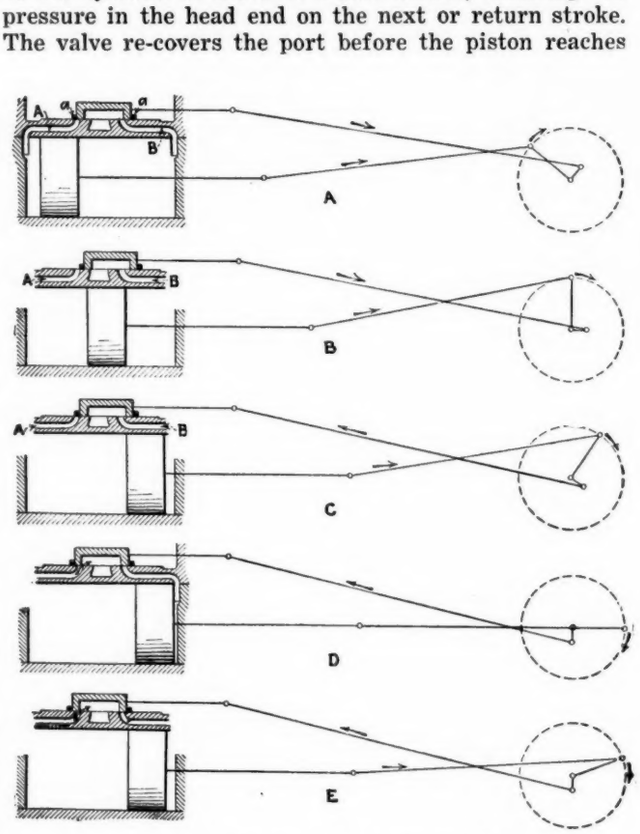


Fig. 3—Results of adding lap to the valve

dead center on the return stroke. The steam left in the cylinder is compressed by the advancing piston, thereby cushioning the moving parts and so eliminating pounding as the piston is brought to rest.

In this discussion it has been assumed that the valve was controlled by a crank. In the actual engine the crank, by enlarging the pin so that it includes the shaft bore, takes the form of an eccentric but still remains in effect a crank arm and pin.

Highest Efficiency from Industrial Steam Plants

By WALTER SLADER

IN ORDER to understand how economies of industrial plants may be increased, we must study the properties of steam. Reference to figures will serve far more clearly than any description to give a correct idea of the way the temperature of steam, total heat of steam, heat of liquid, etc., vary with increasing pressure.

Curve *G* in Fig. 1 shows the change in temperature of saturated steam for the range from one-half pound absolute, or 29-in. vacuum, to 1,000 lb. gage; and like curves *H*, *J*. and *K* show it for steam superheated 100, 200 and 300 deg. F. through the range from zero pounds gage to 1,000 lb. gage.

That part of the range from one-half pound absolute to atmospheric pressure has been marked in terms of the more familiar vacuum pressure, as measured by mercury column and also made to a larger pressure scale than the remaining portion of the diagram that values here may be read more accurately.

In a similar way curve *L*, Fig. 2, shows the variation of B.t.u. total heat, and curve *M* shows that for heat of liquid per pound of saturated steam at various pressures up to 1,000 lb. gage. Another curve *N*, showing the heat remaining in a pound of steam after adiabatic expansion from the higher pressure shown directly above to one-half pound absolute, or practically 29 in. vacuum, is also included. By noting the vertical distance between these curves, a comparison may readily be made between the total heat initially in each pound of this steam at various pressures and the maximum amount that, theoretically, could be converted to work by a 100 per cent efficient turbine or engine.

Brief consideration of what gains can be made through increasing pressure range, assuming theoretically perfect equipment, is worth while, since it shows us the limits beyond which we can never go. A comparison of these theoretical values with those that have thus far been attained or which reasonable estimate indicates that we might attain by the use of known equipment, gives us a measure of our present progress and some indication as to what more may be likely for the future.

Again, the story is most clearly told by reference to a chart, Fig. 3, which shows graphically the data recorded in the table. Each item is given the same distinguishing number in the tabulation and on the diagram.

A group of typical cases of heat use through varying ranges is represented in Fig. 3. For the condensing units the steam required for power to drive auxiliaries is included that the comparison may be more exact. All steam engines are assumed to be driving electric generators, therefore their losses are included in the water rates per kilowatt-hour, and conditions are made comparable for engines and steam turbines. It might theoretically have been preferable to reduce all values to brake horsepower, but this would have introduced more complication, and we are constantly growing to think more in terms of kilowatts.

With both engines and turbines some examples of ordinary machines and some of very high types have been included. They have not been chosen with any purpose of comparing the relative efficiencies of the two kinds of units, but solely to show representative space distributions of energy used and rejected.

Most of these examples represent operating conditions that have been usual in the past or are at present in use; a few represent initial pressures and temperatures higher than at present are ordinary or back pressures higher than are ordinary. The group serves to illustrate the gain that has been and that may be made in useful power output through increasing working range.

Those having to do with back-pressure units show clearly that a worth-while amount of power may be obtained when the exhaust is at a pressure suitable for use in any ordinary manufacturing process.

By joining together the points representing total heat above 32 deg. F. in steam as taken into the engines or turbines chosen for the aforementioned group of examples, we obtain a rough outline curve *P*, Fig. 3, which serves to make clear the extent to which the top range may be varied with increased pressure or superheat. By joining together the points representing the total heat theoretically remaining in a pound of steam when expanded adiabatically from each top range to the designated low-pressure level, curve *Q* is obtained.

The vertical distance between points on curve *P* and the corresponding points on curve *Q* represents in true proportion to the whole the portion of energy theoretically available, or that which could be secured by the use of a 100 per cent perfect engine working within this range.

If, starting from points on the *P* curve, the actual heat energy per pound of steam that is converted into power when used in engine or turbine, is laid off downward, we obtain points that, joined together, form curve *R*.

The general outline of these curves of course has

IN a previous article, "Broadening the Field of Service for Power Plant Engineers," in the March 11 issue it was brought out that an appreciable increase of the use of water power may be expected. Efficient means of utilizing water and steam were reviewed. An economy of heat utilization far exceeding the large condensing station is possible in a small industrial plant under conditions here outlined. Co-ordinating industrial processes to bring about this result requires a special type of engineering training.

no special meaning, this being dependent solely on the arrangement of the examples chosen, but the points have been thus joined together that the relative size and proportion of the areas representing energy theoretically available, energy used and energy that must be rejected to waste may be more clearly seen.

Distances between curves *P* and *R* show the part of total energy actually realized as power. Distances between curves *R* and *Q* show the part rejected through inefficiency of engine or turbine. Distances between curve *Q* and the base show total heat above 32 deg. F. that must be rejected by a perfect engine, while similar distances between curve *R* and the base show the total that actually is thus rejected. In the Northern New England section, from which this is written, the average yearly temperature is about 45 deg. F., which corresponds to 13 B.t.u. on the diagram. A line has been drawn through this point forming curve *S*. Heat between the top level and that corresponding to curve *S* has, as a year's average, to be put into each pound of steam.

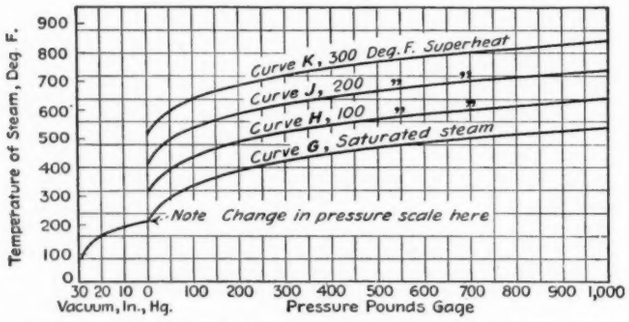


Fig. 1—Relation between pressure and temperature of steam

For the purpose of illustration, we assume a uniform feed-water temperature of 210 deg. F., which could be obtained in any of these cases by the use of exhaust from the main unit or auxiliaries, but which is considerably higher than the average actually found in most plants. The heat that must be put into each pound of steam by the use of new fuel is shown by the distance between points on curve *P*, Fig. 3, and curve *T* located at the 178 B.t.u. level. The ratio $\frac{PR}{PT}$ shows the percentage of total energy realized as power compared to the total delivered to each pound of steam from fuel used.

Conclusions from the Figures

It will be noted that in order to make any great further gain in useful power realized through increase of working range, either the pressure and temperature of steam must be quite markedly increased, which beyond a limited extent will require much heavier and more expensive construction, or the use of some other working fluid such as mercury vapor or carbon-disulphide along with steam must be considered. Such systems would be both more complex and more expensive to maintain.

Mr. Orrok, in a recent paper before the A.S.M.E., called attention to the fact that by the use of a system of feed-water heating in steps, using several heaters and bled steam at a series of increasing pressures from power units, the feed water may be returned to boilers at temperatures much nearer that of the heat in water at the high-pressure level than is now usual.

This in effect raises the level of curve *T*, Fig. 3, saves heat otherwise wasted and considerably improves the efficiency of operation over that possible with the more usual Rankine cycle. This gain is made, of course, at the expense of considerable increase in complexity and cost of the system. The theoretical limit of such heat return is shown by the "heat of liquid" line curve *M* in Fig. 2, and it is clear that, while this possible saving is important and considerable, the rejected heat area still remains a large one.

Bleeding Steam for Heating Feed Water Cannot Greatly Increase Efficiency

The fact that should be particularly noted is that the area of rejected heat changes only a comparatively small degree as shown by the total areas under curves *R* and *Q*, Fig. 3, regardless of whether the base line is taken at that of 32 deg. F. corresponding to zero heat values from standard steam tables or at ordinary feed-water temperatures, as curves *T* for 210 deg., Fig. 3, or at the new higher level that has been suggested and has its limit at the "heat of liquid" line. That is, in any event with our present methods of generating power a very large percentage of the total heat delivered from fuel to steam at throttle must be rejected to waste unless means can be found to use this heat in other ways.

Many such means have been found and are more or less in use. I believe that still other such means can be found and that the use of such available means should be greatly increased. The portion of heat that now ordinarily goes to waste is such a large one that, if only a reasonable-sized fraction of this part can be saved it will represent a great gain.

It takes but a relatively small added amount of heat to increase the pressure and temperature of steam greatly. The power-producing units make good use of this heat at the high range, but reject enormous quantities at the lower levels.

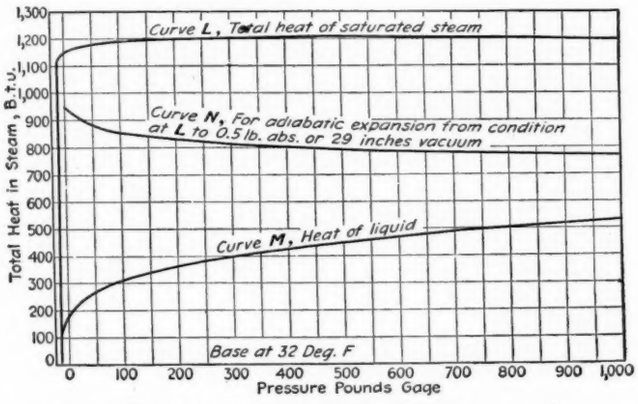


Fig. 2—Total heat of saturated steam is maximum near 500 pounds

By shortening somewhat their range of work, a good amount of power can be obtained and still leave the steam from their exhausts fully suitable for the industrial processes. The problem then is to so harmonize the requirements of power production and manufacturing processes that full use may be made of this heat through the widest practicable range.

One of the earliest applications of this idea was,

naturally, the use of exhaust steam from a non-condensing engine at atmospheric pressure or slightly above to heat water. Then a little higher pressure, from the exhaust of a non-condensing engine or the receiver of a compound condensing engine, was used to heat buildings and to furnish steam to slashing,

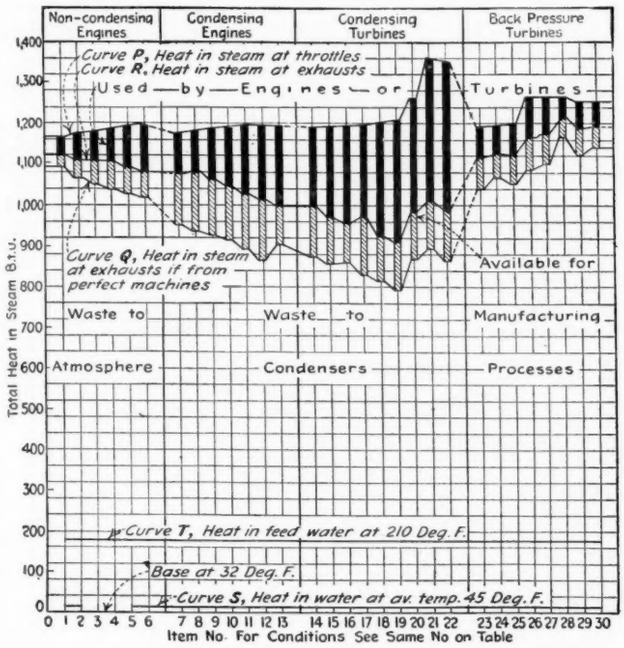

Non-condensing Engines
Condensing Engines
Condensing Turbines
Back Pressure Turbines
Curve P, Heat in steam at throttles
Curve R, Heat in steam at exhausts
Used by Engines or Turbines
Curve Q, Heat in steam at exhausts if from perfect machines
Available for
Waste to Atmosphere
Waste to Condensers
Manufacturing Processes
Curve T, Heat in feed water at 210 Deg. F.
Base at 32 Deg. F.
Curve S, Heat in water at av. temp. 45 Deg. F.
Total Heat in Steam B.t.u.
Item No. For Conditions See Same No on Table

Fig. 3—Heat utilized in power a small proportion of that rejected at exhaust

bleaching, dyeing, other textile finishing processes and the drying of paper.

Heating demands not only extend over merely a portion of the year, but are intermittent in different plants to a widely varying degree. A simple non-condensing equipment, with its exhaust used for heating the plant, will be highly efficient during a part of the time and may give by much the cheapest over-all fuel cost; on the other hand, it may be so wasteful during the remaining periods as to more than swallow up its earnings as compared with a condensing unit. Its success or failure will depend upon proper adaptation to the special conditions.

Most process uses are variable in their steam demands, some of them extremely so; the extent of these variations have often not been understood. Such demands can usually be properly met, while making full use of exhaust steam, but in order to do so, they must be clearly foreseen and provided for.

The old-type compound engine is not well suited to meet the requirements of wide fluctuations in demand for such heating or process steam bled from its receiver.

When the steam turbine came into general use, it was soon found that this machine, either non-condensing against back pressure or bleeding from an intermediate stage, was better adapted to meet the varied requirements for economically furnishing both power and steam for process work.

The non-condensing turbine, except as a unit in a large plant operating in parallel with condensing units, is subject to many of the same limitations as the non-condensing engine. The application of both this and the bleeder type was at first mostly or entirely limited to relatively low pressures, 5 to 20 lb. gage.

Until recently, bleeding at high pressures had not been worked out. Some time ago, in our engineering practice, we saw the need for better means of taking care of these conditions, and we believe we can rightly claim a considerable part in pointing out these needs and devising the general details of a group of machines to better meet them. In so bringing the attention of standard turbine manufacturers to the advantage of such equipment, there has resulted, as the product from several builders, both non-condensing and extraction type turbines suitable for producing

TABLE OF DATA FOR STEAM ENGINE AND TURBINE PERFORMANCE CURVES, AS ILLUSTRATED IN FIG. 3

Item Number	Kind of Unit	Type of Prime Mover	Steam Conditions	Total Initital Heat, H_1	Total Theoretical Final Heat, H_2	H_1-H_2 Theoretical	Steam Rate per I.Hp.-Hr.	Steam Rate per Kw.-Hr.	B.t.u. used per Lb., I.Hp.-Hr.	B.t.u. used per Lb., Kw.-Hr.	Ratio I.Hp.-Hr. Actual / Theoretical	Ratio Kw.-Hr. Actual / Theoretical
1	Non-condensing Engines	High type Corliss	4 lb. abs.-sat. to ats.	1,169	1,096	73	46.0	73.2	55.4	46.4	75.7%	63.7%
2		High type Corliss	65 lb. abs.-sat. to ats.	1,178	1,070	108	31.3	50.3	81.0	68.0	75.0	63.0
3		Four slide valve	90 lb. abs.-sat. to ats.	1,184	1,054	130	28.3	45.5	90.4	75.3	69.5	57.7
4		Corliss engine	115 lb. abs.-sat. to ats.	1,189	1,040	149	26.25	42.4	97.3	80.7	65.2	54.2
5		Cross-compound	140 lb. abs.-sat. to ats.	1,192	1,030	162	22.0	35.5	116.0	96.0	71.6	59.0
6		Unaflow	165 lb. abs.-sat. to ats.	1,195	1,020	175	18.7	29.8	136.0	114.3	77.7	65.4
7	Condensing Engines	Simple Corliss	65 lb. abs.-sat. to 2 lb. abs.	1,178	951	227	19.6	32.7	130.0	104.4	58.0	46.0
8		Four slide valve	90 lb. abs.-sat. to 2 lb. abs.	1,184	936	248	21.2	35.3	120.0	96.7	48.8	39.1
9		Unaflow	115 lb. abs.-sat. to 2 lb. abs.	1,189	925	264	16.3	27.8	156.0	128.0	59.0	48.4
10		Corliss-compound	140 lb. abs.-sat. to 2 lb. abs.	1,192	915	277	14.0	23.5	183.0	145.0	66.0	52.4
11		Corliss-compound	165 lb. abs.-sat. to 1.5 lb. abs.	1,195	893	302	12.7	20.3	201.0	167.0	66.3	55.5
12		Triple-expansion	215 lb. abs.-sat. to 1.5 lb. abs.	1,199	861	338	11.75	18.5	208.0	184.5	64.0	54.6
13		Old 5,000-kw. interborough	190 lb. abs.-sat. to 2.25 lb. abs.	1,198	905	293	....	17.1	.....	199.0	....	68.2
14	Condensing Turbines	Standard-2,500 kw.	165 lb. abs.-sat. to 1 lb. abs.	1,195	873	322	....	17.4	.....	196.0	....	61.0
15		Standard-5,000 kw.	215 lb. abs.-sat. to 1 lb. abs.	1,199	860	339	....	15.2	.....	225.0	....	66.3
16		Multi-stage-5,000 kw.	215 lb. abs.-sat. to 1 lb. abs.	1,199	860	339	....	14.2	.....	240.0	....	70.9
17		Standard-5,000 kw.	215 lb. abs.-sat. to 0.5 abs.	1,199	830	369	....	14.9	.....	229.0	....	62.2
18		20,000-kw. unit	315 lb. abs.-sat. to 0.5 abs.	1,204	814	390	....	12.2	.....	280.0	....	71.8
19		Large unit	515 lb. abs.-sat. to 0.5 abs.	1,210	792	418	....	11.4	.....	299.0	....	71.6
20		Multi-stage-5,000 kw.	215 lb. abs.-500 deg. F. to 0.5 abs.	1,265	869	396	....	12.2	.....	280.0	....	70.8
21		Large unit	315 lb. abs.-700 deg. F. to 0.5 abs.	1,360	895	465	....	9.75	.....	350.0	....	75.3
22		Large unit	515 lb. abs.-700 deg. F. to 0.5 abs.	1,353	865	488	....	9.40	....	362.0	....	74.3
23	Back Pressure Turbines	Standard machine	165 lb. abs.-sat. to 20 lb. abs.	1,195	1,040	155	....	42.4	.....	80.6	....	52.0
24		Standard machine	165 lb. abs.-sat. to 30 lb. abs.	1,195	1,067	128	....	51.3	.....	66.4	....	52.0
25		Standard machine	215 lb. abs.-sat. to 30 lb. abs.	1,199	1,052	147	....	44.6	.....	76.5	....	52.0
26		Standard machine	190 lb. abs.-500 deg. F. to 20 lb. abs.	1,268	1,084	184	....	33.8	.....	101.0	....	55.0
27		Standard machine	215 lb. abs.-500 deg. F. to 30 lb. abs.	1,265	1,103	162	....	38.3	.....	89.0	....	55.0
28		Special machine	215 lb. abs.-500 deg. F. to 75 lb. abs.	1,265	1,172	93	....	67.0	.....	51.0	....	55.0
29		Special machine	365 lb. abs.-500 deg. F. to 75 lb. abs.	1,254	1,123	131	....	50.0	.....	68.2	....	52.0
30		Special machine	365 lb. abs.-500 deg. F. to 95 lb. abs.	1,254	1,140	114	....	57.6	.....	59.3	....	52.0

Note: Steam rates of items 7 to 22 allow for power to drive condenser.

power and delivering steam in quantities as required for industrial uses at any pressure or pressures up to 125 lb. gage.

I have had occasion to estimate the cost, and performance of several such equipments carefully, and often they show splendid earnings.

One such unit, now about to go into operation, provides for the extraction of steam at 60 lb. gage for use in certain processes, and the exhaust of the remainder at 10 lb. gage for other plant uses. This machine can supply power in quantities according to steam demands up to 600 kw. Sizes can be made as required and the details of extraction and exhaust pressures, one or both, varied through any reasonable range.

These machines may sometimes be located in a main turbine room alongside other units, or sometimes best in spare space in the boiler room; or yet again in some distant part of the plant close to the point of use for their exhaust. In the smaller sizes they can be made so as to require little operating attention beyond that necessary for starting and stopping. They usually operate in parallel with other plant units or with central-station service. Special governors and appropriate relay mechanism provide full protection against possible emergency conditions.

Units of the type mentioned are in reality nothing other than high-pressure, power-producing, reducing valves, and this seems a good though somewhat lengthy name for them.

Turning to the engine or steam-turbine room, the diagram, Fig. 2, of the article "The Broadening Field of Service for the Power Plant Engineer" in the March 11 issue, enables you to see the part that is delivered to the switchboard or busbars and the part that finally remains when it arrives at the machine pulley that is the real point of use.

You will, perhaps, be surprised to see how small a percentage this is at best, for although we know these facts, we are inclined to overlook or forget them.

Large Savings Possible in Existing Industrial Plants

Small as this useful portion is, as obtained from the best plants, it is particularly worth while to note how much less it is from the ordinary and poor ones. While we shall make further improvement over that which we now obtain from these best plants, the opportunity for large gains lies much more in bringing the ordinary and poor ones up to the level of the good ones.

This is really the largest and most helpful field for energy saving. The difference between 8.5 per cent and 4.7 per cent energy finally realized represents a saving of 80 per cent of that made useful under the poor condition, while the difference between 6 per cent and 3 per cent represents a doubling of useful output.

Results will be obtained in proportion as there is a realization of this possibility, that it is worth while in dollars saved to the owner, and that it can be brought about only through the continuous work of specially trained men.

Fig. 3 is intended to make clear the reason for, and enforce the argument regarding, the necessity of combining our power and manufacturing steam uses to make energy and dollar savings. This shows some of the reasons why a pound of steam should not go to a condenser when it can be used about the plant. Now that we have the high back-pressure turbine, there is less reason for this than ever before.

Steam Boilers

A Catechism by Warren Hilleary Served in Small Doses and Extending Over a Number of Issues

90. The blowoff from a certain boiler is piped to a cistern or hotwell, and the blowoff pipe must enter this cistern near the top. Would you extend the blowoff pipe below the water level at the bottom of the cistern, or would you carry it downward only a few inches?

The pipe should have an elbow and a short nipple extending downward only a few inches. In no case should the discharge be below the water level in the cistern. If the blowoff pipe were extended below the water, there would occur so great an agitation of the water as to be dangerous. Moreover, it is advisable to avoid unnecessary back pressure on the blowoff line.

91. On making an inspection of a water-tube boiler under pressure, suppose you find that some of the tubes are bagged considerably and there appears to be water issuing from a crack in the apex of a bag. Would you demand that the boiler be shut down immediately, provided the regular time for closing the boiler down was one hour later than the time you make your discovery, or would you allow the boiler to run that one hour?

The fire should immediately be drawn, all drafts left wide open and steam pressure reduced as rapidly as possible by admitting feed water. The bagged tube should then be removed.

92. A boiler owner complains of his fuel bills. The inspector finds there is no feed-water heater and the feed water is entering the boiler at a temperature of about 60 deg. What saving in fuel will take place if the owner installs a heater capable of raising the feed-water temperature to 200 deg. F.

About 12 per cent theoretically, the basis being that for each 12 deg. increase in feed-water temperature there is an approximate saving of 1 per cent in fuel consumption. Actually, the saving would probably be nearer 9 per cent than 12 per cent.

93. (a) What is single pitch? (b) Double pitch? (c) Longitudinal pitch? (d) Circumferential pitch?

(a) The closest pitch required for any part of a seam is single pitch; (b) double pitch, in any seam, is twice as great as the single pitch in the same seam; (c) the distance from center to center of rivets in any row that is parallel with the boiler or drum length; (d) the distance from center to center of the outside heads of the girth seam rivets.

94. A boiler plate is ¾-in. thick. (a) What size rivet hole should be used in the longitudinal double-riveted seams? (b) What size rivet should be used?

(a) Thirteen-sixteenths inch. (b) Three-quarter inch.

95. Is it customary to measure the distance from the center of a rivet hole to the edge of the sheet or from the edge of the rivet hole to the edge of the sheet, and why?

From the edge of the rivet hole to the edge of the sheet because there must be a minimum width of this ligament and to measure from the center of the hole to the edge of the sheet would vary the width of the ligament with varying sizes of hole.

96. How far above the top row of tubes of a horizontal tubular boiler should the bottom of the gage glass be located?

Two inches.

Figuring Centrifugal Pump Characteristics from Those at Known Speed

By Frederick C. Evans*

In the modern steam power plant the variable-speed centrifugal pump can be used to advantage in many applications. Where variable discharge is required, as is the case with boiler feed or condenser circulating pumps, variable-speed control results in greater operating economy than by throttling the discharge.

In preliminary calculations, before the pumps have been purchased, where no test results are available,

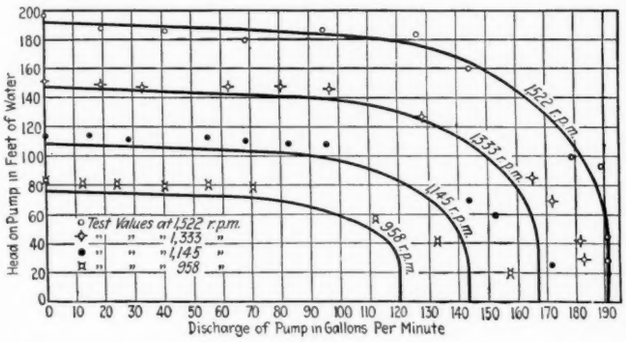

Head on Pump in Feet of Water
Discharge of Pump in Gallons Per Minute
1522 r.p.m.
1333 r.p.m.
1145 r.p.m.
958 r.p.m.
Test Values at 1,522 r.p.m.
" " " 1,333 "
" " " 1,145 "
" " " 958 "

Fig. 1—Discharge heads computed for lower speeds check practically within 10 per cent of tests

it is often desirable to predict the characteristics at various speeds. Questions arise, such as, For what speed range must the driving unit be designed in order that all conditions of service may be met? Knowing the characteristics of the driving unit, what power will be required to drive the set under various conditions of service?

Calculations to predict these variable speed characteristics are based on the following laws of centrifugal-pump operation:

1. The discharge of the pump varies as the speed.
2. The head varies as the square of the speed.
3. The water or delivered horsepower varies as the cube of the speed.

Characteristic Curves at Maximum Speed Are Determined from Catalogs or Handbooks

The character of the service will fix the type of "head-discharge" curve required, whether, for example, it is "flat" or "steep." With the aid of catalogs or published articles reasonable characteristics of the type required may be found for one speed. It is usually most convenient to determine the characteristics for the maximum speed desired. The head required and the amount of water handled will give an idea of the maximum efficiency that may be expected.

The head, the discharge, the efficiency and the water horsepower can then be tabulated for several specific points at this speed, these being taken over a desired range of the characteristics. Then for each, a corresponding point can be determined on the characteristic curves at any other desired speed. These computations would be made as follows:

$$q_2 = \frac{n_2}{n} q_1;\ h_2 = \left(\frac{n_2}{n_1}\right)^2 h_1 \text{ and } p_2 = \left(\frac{n_2}{n_1}\right)^3 p_1$$

where

n_1 = The speed at which the characteristics are known;
n_2 = The speed at which it is desired to estimate the characteristics;
q_1 = The known discharge corresponding to n_1;
q_2 = The desired discharge corresponding to n_2;
h_1 = The known head corresponding to n_1;
h_2 = The desired head corresponding to n_2;
p_1 = The known water horespower corresponding to n_1;
p_2 = The desired water horsepower corresponding to n_2.

For example, suppose a certain type of pump will deliver 30,000 gal. per min. against a head of 38 ft. of water when running at 850 r.p.m. The brake horsepower[1] under these conditions amounts to 351. What would be a corresponding point on the characteristic curves at 400 r.p.m.? The discharge at 400 r.p.m. is

$$30{,}000 \times \frac{400}{850} = 14{,}110 \textit{ gal. per min.}$$

The corresponding head at 400 r.p.m. is

$$38 \times \left(\frac{400}{850}\right)^2 = 8.4 \textit{ ft. of water}$$

The corresponding brake horsepower is

$$351 \times \left(\frac{400}{850}\right)^3 = 36.5$$

Other points could likewise be computed for 400 r.p.m.

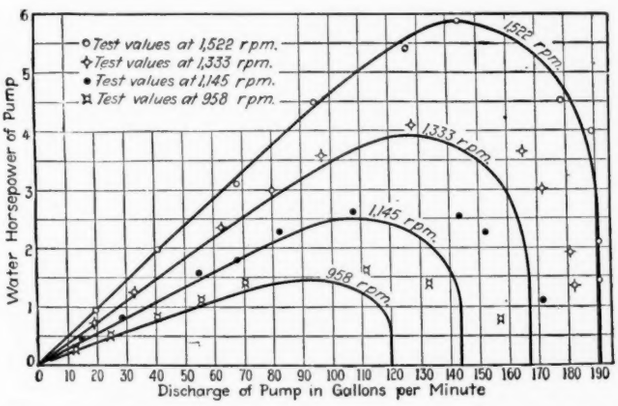


Fig. 2—Calculated delivered power at lower speeds checks within nearly 8 per cent of tests

from other known values at 850 r.p.m. and the characteristic curves at the lower speed determined.

The question now is, How closely true are the three laws on which these calculations are based? Because, of course, they must be reasonably close or the conclusions that would be drawn from a study of variable speed operation would not be dependable.

In an effort to satisfy himself on this point, the author made an investigation based on data of a 2½-in. two-stage centrifugal pump rated at 210 ft. head, 1,600 r.p.m., which was tested at various speeds.

The head-discharge curve at 1,522 r.p.m., shown in Fig. 1, was assumed, and from this the head-discharge curves for speeds of 1,333, 1,145 and 958 r.p.m. were calculated. Fig. 1 shows the comparison of the calcu-

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[1]The brake horsepower is the horsepower input to the pump. See last paragraph for reason for using brake horsepower instead of water horsepower.

lated head-discharge curves at these lower speeds and the test values at these same speeds. It is apparent that the agreement is reasonably close over the major portion of the curves, but at the lower speeds the head does not actually drop off as suddenly at the higher discharges as the computations would indicate. However, this is beyond the efficient range of the operation of the pump and for this reason will not enter into most problems.

Since the water horsepower or output of the pump is proportional to the product of the head and the discharge, a study of Fig. 1 would indicate that, as the speed dropped, the water horsepower at the higher discharges would be higher than calculated. Fig. 2 shows that this is true, although the error again becomes serious only when the pump is operated beyond the efficient range.

In making calculations it is usually necessary to work with the water horsepower, or pump output. That the brake horsepower or input rather than the water horsepower or output varies as the cube of the speed is usually assumed. Although this introduces a further error, as explained on pages 96 and 97 of the first edition of "Centrifugal Pumps," by R. L. Daugherty, output must be generally used in making calculations because the variation of pump efficiency with the speed is not known when a preliminary study is required.

Condenser Unit Heat Transfer Obtained from Chart

BY H. O. MICHAEL

This is a chart with which the operating engineer may very quickly learn the condition of his condenser. It is based on the general principle that all central stations and other well-regulated plants maintain a fairly constant steam pressure and temperature. Because of this, the B.t.u. rejected to the condensers per pound of steam is nearly constant, varying slightly with the vacuum.

A separate chart must be made for each different condenser, as the cooling surface varies with the design and size of each. One chart can, however, be used with any number of duplicate condensers.

To begin, steam flow, as obtained from water-rate curves of the turbine, is plotted against B.t.u. lost to circulating water, for several condensate temperatures within the operating range. The scale for B.t.u. transmitted per hour per square foot of cooling surface is constructed by dividing the B.t.u. lost at any point by the square feet of cooling surface of the condenser in question.

The mean temperature difference is found from the chart at the right, which was redrawn from one printed in *Power*. Finally, the B.t.u. transmitted per square foot per hour for different conditions is divided by figures representing the practical range of mean temperature difference, giving data for plotting the scale at the top, which is B.t.u. transmitted per square foot per hour per degree of mean temperature difference. This latter figure indicates the condenser performance.

To use the chart, observe the load on the unit, and the condensate, injection and discharge temperatures. Then knowing the steam consumption at the observed load, and finding the mean temperature difference from the curves at the right, begin at the bottom, follow as shown by the broken line, first vertically to the condensate temperature line, then horizontally to the mean temperature line, and finally vertically to the top.

As an example, we will assume averaged data with 18,087 kw. load on the unit. At this load the turbine uses 220,000 lb. of steam per hour. The data and calculations in detail appear in the table.

Beginning at the bottom of the chart, on the steam-flow scale, trace vertically upward from the 220,000 mark to the 89 deg. condensate line. Use 90, as the lines are so close. From this intersection trace horizontally to the mean temperature line at the right. The mean temperature difference should be found from the small chart at the right of the figure.

MEAN TEMPERATURE DIFFERENCE FROM CHART

To find the mean temperature difference, item 28 in the table, the condensate minus injection temperatures is, taken from items 8 and 9, 89 — 39 = 50 deg. F. The condensate minus discharge temperatures is from items 8 and 10; 89 — 57 = 32 deg. Beginning with 50 in the small curves to the right of the figure and moving to 32 deg. as the curves indicate, the mean temperature difference is 40.5 deg., shown by the dotted line.

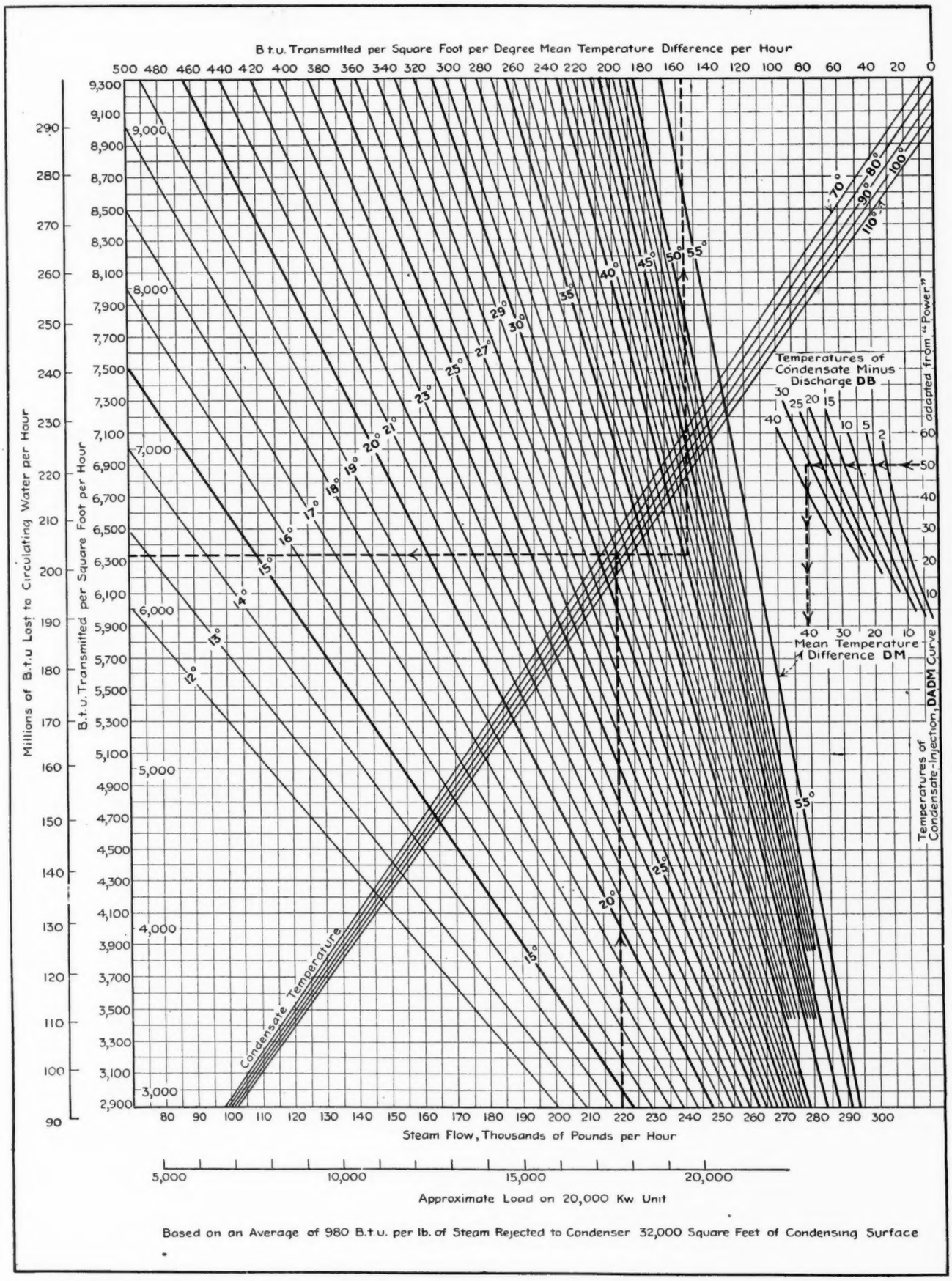
From this last intersection at the right of the main figure, move vertically upward to the scale at the top, where the B.t.u. transmitted per degree per hour may be read directly. In this case it is found to be 157 B.t.u. per sq.ft. per deg. per hour.

Should it be desirable to know the B.t.u. lost to the circulating water, or B.t.u. transmitted per square foot of cooling surface, trace horizontally to the left from the intersection of the 90-deg. condensate line and the vertical line from the 220,000 point on the steam-flow scale, and read directly on the left margin of the chart. We find this to be 203 million B.t.u. lost to the circulating water per hour, and 6,300 B.t.u. transmitted per square foot of cooling surface per hour.

TABLE OF CONDENSER TEST

ITEM	
1 Average load, kw. (observed)	18,087
2 Steam pressure at throttle (constant)	190
3 Steam temperature 499 deg. F. superheat (constant)	115
4 Barometer, in Hg. (observed)	30.35
5 Vacuum in. Hg. (observed)	29.09
6 Absolute pressure, in. Hg. (from items 4 and 5)	1.26
7 Temperature corresponding to vacuum	87
8 Temperature of condensate (observed)	89
9 Temperature of injection (observed)	39
10 Temperature of discharge (observed)	57
11 Temperature rise of injection (from items 9 and 10)	18
12 Terminal difference, deg. (from items 8 and 11)	32
13 Total pounds of steam entering turbine per hour (from item 2)	220,000
14 B.t.u. per pound of steam at throttle (constant,—from items 13, 2 and 3)	1,265
15 B.t.u. per pound of steam entering condenser after adiabatic expansion (from Mollier diagram)	878
16 B.t.u. per pound available in turbine (item 14 minus item 15)	387
17 B.t.u. used per pound of steam with a Rankine efficiency of 73.6 percent. (item 16 x 0.736)	285
18 B.t.u. rejected to condenser per pound of steam (items 16 minus 17 plus 15)	980
19 Total B.t.u. rejected to condenser (from items 18 and 13)	215,600,000
20 Total B.t.u. retained by condensate (from items 8 and 13)	12,540,000
21 B.t.u. to raise leakage to condensate temperature	(negligible)
22 B.t.u. lost to injection water per hour (items 20 minus 19; also given on chart)	203,060,000
23 "Q" injection water, pounds per hour (from items 22 and 11)	11,281,000
24 "Q" injection water, pounds per minute (from item 23)	188,300
25 "Q" injection water, gallons per minute (from item 24)	22,580
26 Water to steam ratio (pounds) (from items 23 and 13)	50.66
28 Mean difference in temperature (from items 8, 9, 10 and curves at right)	40.5
29 B.t.u. transmitted per hour per sq.ft. (from items 13, 8, and chart)	63.60
30 B.t.u. transmitted per hour per sq. ft. per degree mean temperature difference (from items 29 and 28, also from chart)	157

When the ammonia vapor is condensed in the condenser, that portion of the lubricating oil vapor that has its condensation temperature higher than the cooling water temperature will liquefy. This oil passes into the receiver along with the ammonia.


B.t.u. Transmitted per Square Foot per Degree Mean Temperature Difference per Hour
Millions of B.t.u Lost to Circulating Water per Hour
B.t.u. Transmitted per Square Foot per Hour
Condensate Temperature
Steam Flow, Thousands of Pounds per Hour
Approximate Load on 20,000 Kw Unit
Temperatures of Condensate Minus Discharge DB
Mean Temperature Difference DM
Temperatures of Condensate-Injection, DADM Curve adapted from "Power"
Based on an Average of 980 B.t.u. per lb. of Steam Rejected to Condenser 32,000 Square Feet of Condensing Surface

Unit heat transfer found from total steam per hour, condensate, injection and discharge temperatures

Operating Instructions for Large Turbines

Operating engineers, regardless of the types of turbines which they handle, may find useful ideas by reading these instructions. They refer primarily to the present types of large Westinghouse turbines, such as that described in the May 15, 1923, issue, but are almost equally applicable to those of earlier manufacture. This article is not put forward as authoritative for any special condition, the manufacturer being the proper source for full particulars in regard to any individual machine.

THE following may be regarded as good general rules of procedure in starting up and getting the turbine under way, from the instructions of the Westinghouse Electric & Manufacturing Co.

1. Establish water circulation through the condenser.

2. Start auxiliary oil pump. Be sure it is under proper control of the regulator.

3. Depress the collapsible link between the governor and steam chest and be assured that the whole valve gear works freely.

4. Trip the safety stop by hand and be sure the mechanism of the automatic throttle functions freely and properly.

5. See that the steam line is free of water and that all turbine drains are open.

6. Quickly open the throttle valve a sufficient amount to start the spindle revolving, immediately closing it, and then opening it enough to maintain the spindle operating at from 100 to 200 revolutions per minute.

7. Immediately following the foregoing, apply gland steam in sufficient quantities so that there is a moderate amount blowing outward into the engine room from each gland.

8. Then start the air pump or ejectors.

9. Keep the turbine running at this speed until it is thoroughly warmed with this flow of steam, consuming, under ordinary conditions, about thirty minutes. As during this time the vacuum will be rising, the gland-sealing steam must be increased as required. Close whatever drains may be open. The vacuum should not be allowed to reach more than 20 in. to 25 in. in order that the inlet pressure may not be less than 15 lb. gage and there is ample flow of steam through the turbine. If necessary, check further rise by admitting a small flow of air into the condenser.

10. Start the condensate pump as soon as the throttling valve has been opened a second time and there is some flow of condensate. To operate the condensate pump without water endangers it because of possible contact with the sealing rings. It may be necessary to operate the condensate pump for a brief time before this in the event of the condenser having become flooded for any reason.

11. Bring the turbine slowly up to speed, taking about fifteen minutes in the operation. When the turbine is somewhere between three-quarters and full speed, turn on gland water and close off the gland-sealing steam.

12. Obtain assurance that the governor properly controls the turbine with a wide-open throttle, full steam pressure on the line, full vacuum in condenser and without field on the generator.

13. Synchronize the machine as quickly as possible and apply some 2,000 or 3,000 kw. of load, and immediately thereafter permit the vacuum to reach its maximum.

14. Before the turbine carries any material load, open the valve connecting the leak-off from No. 2 gland to the low-temperature heater zone.

Stage Feed-Water Heaters

During all the starting operations the condensate pump may be pumping water through the heaters and the turbine draining and bleeding into them. Unless it is necessary to do some work on the heater connections, the valves in connection therewith do not need to be changed from day to day in stopping and starting. There must at all times be open drains from the heater zones either through the heaters themselves or bypassing them if they are disconnected.

Admit water to oil cooler.

Inspections During Operation

During operation, observe the following:

1. See that all oil pressures in the lubricating system are properly maintained.

The relief valve adjacent to the hydraulic relay should be in adjustment so that from 50 to 55 lb. pressure is maintained on the governor oil system. The pressure of oil in the bearing system should approximate 5 lb. at the turbine axis. This latter pressure is dependent only on the resistance to flow through the bearing, etc., setting of auxiliary oil-pump regulator, the capacity of the main pump and the temperature and viscosity of the oil. No adjustment of the aforementioned high-pressure relief valve can appreciably change the bearing-oil pressure. Lack of bearing-oil pressure can be due only to excessive clearances in bearings, oil bypassing at some place to the reservoir or insufficient capacity of the main oil pump.

Each bearing should raise the oil temperature 15 to 20 deg. F. The oil should not leave the bearings in excess of 140 deg. F. Adjust the water supply to the oil cooler to approximate this temperature.

2. The governor dashpot should have its needle valve adjusted to be as wide open as possible consistent with stability, so the governor may move quickly in response to changes of load.

3. Observe and maintain a record of pressures and temperatures, particularly stage pressures.

4. Various governor control linkages, etc., should be oiled at proper intervals.

5. See that the auxiliary oil-pump regulator is in condition by spinning the weights. At intervals push down the weights and observe whether the auxiliary pump starts up freely.

6. Regard as important any disturbance or vibration,

no matter how trivial. In the event of serious vibration, the machine should be taken off the line and the cause investigated.

Rules for Shutting Down

The following may be regarded as a good general rule of procedure in shutting down:

1. Close the throttle as soon as signal is received that main switch is open.
2. Break vacuum.
3. Shut down air pump or ejectors.
4. Shut down condensate pump.
5. Shut off gland-water supply.
6. Close No. 2 gland leak-off (at exhaust end) from low-temperature heater zone.
7. See that the auxiliary oil pump starts up automatically as soon as the main oil pump ceases to maintain proper oil pressure on the bearing system.
8. Shut down the auxiliary oil pump after the turbine has come to rest.

Note that in shutting down, it is a good plan, say once a month, to close the throttle automatically by tripping the latches connected with the stop governor and admitting high-pressure oil thereto as described by special instructions, thereby testing out the complete chain of automatic stop mechanisms. This latter method of closing the throttle should not be employed so frequently as to cause rapid wear of the auto stop weight and lever.

9. At intervals, when shutting down, observe after the generator circuit breaker has been opened, whether the governor and valve gear control the turbine with an effective full throttle opening and high vacuum. Its failure to control under this condition would indicate leakage of one or other of the control valves or that the governor linkage is in serious maladjustment. This should be rectified before the turbine is again operated.

Precautions That Should Be Observed

1. Avoid at all times steam leaking through the turbine with spindle at rest.
2. Do not operate the air pump with the spindle at rest, and likewise avoid at all times air being drawn through the glands.
3. Avoid the applying of sealing steam to the gland when not necessary.
4. The turbine should not be operated at full speed with no load. When so operating, the steam flow through the turbine is small and the pressure difference over the various labyrinths correspondingly small, so that should there be any rubbing and local heating, there would be insufficient flow of steam to carry away the generated heat; a slight rub may therefore develop into serious trouble. This is particularly true in the early life of the turbine, when the castings are unseasoned and a slight distortion likely to occur. In starting up for the first time and there is confusion and delay in putting the machine on the system because of the checking up of electrical connections, etc., the service engineer should slow the turbine down until he has assurance that all is in readiness for phasing in. Oppose any requests to run for long periods at no load.
5. In starting up with the machine cold, open the throttle valve quickly enough to start the spindle revolving slowly. The purpose of this is to avoid blowing steam through the turbine with the spindle at rest.
6. Avoid as much as possible the undue heating of the low-pressure portions of the turbine beyond what is required for operation under vacuum. This is to prevent waste of fuel and unnecessary stresses due to expansion of parts of the exhaust chamber. Therefore, apply gland-sealing steam and start the air pump immediately after opening the throttle valve, so that the vacuum will be rising while the turbine is slowly revolving, the first opening of the throttle valve adding no appreciable heat to the low-pressure portions. The warming-up process of the turbine is therefore accomplished while the spindle is revolving slowly, bringing it to speed gradually after it is warmed.

See that the entire oiling system is scrupulously clean. Oil from 125 to 175 seconds Universal Saybolt viscosity at 100 deg. F. will be found satisfactory. Governor dashpot and throttle-valve dashpot should be filled with the same oil. Employ only mineral oil free from animal or vegetable adulterants. Avoid oil that has a tendency to emulsify, form sludge or turn acid. Be careful that water does not become accidentally mixed with the oil.

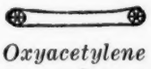
Stage-Pressure Records

By maintaining a record of the various stage pressures in the turbine, any blade derangement may become evident before serious injury results. The manufacturer will supply a curve of the various stage pressures plotted against load. However, this may not be quite accurate. Such a plotting made by the manufacturer is based on calculations of the various turbine-element efficiencies and on the prescribed blade areas, all of which would vary slightly in the actual machines. It is a good plan, after the valves have been properly set, for the service engineer to standardize the gages and make a plotting of stage pressures against load while the turbine is in good condition, making the observations with normal vacuum. He will plot the following observations against load: (1) Primary inlet steam pressure; (2) secondary inlet steam pressure; (3) tertiary inlet steam pressure; (4) impulse wheel chamber pressure (this is the first gage on the side of the exhaust chamber called 110 lb. zone); (5) 55-lb. heater zone pressure; (6) 200 deg. F. heater zone pressure.

With the foregoing curves carefully drawn, check up from day to day to ascertain if the various pressures at the loads being carried correspond with the curves, any change in pressure indicating a change that should be at once sensible of explanation. If injury to blading is indicated, the machine should be taken off the line and an examination made immediately.

Variations in vacuum will slightly affect the stage pressures.

The foregoing method of obtaining knowledge that the turbine blading is in good condition is not common knowledge. Service engineers should commend this practice to the station operating engineers.



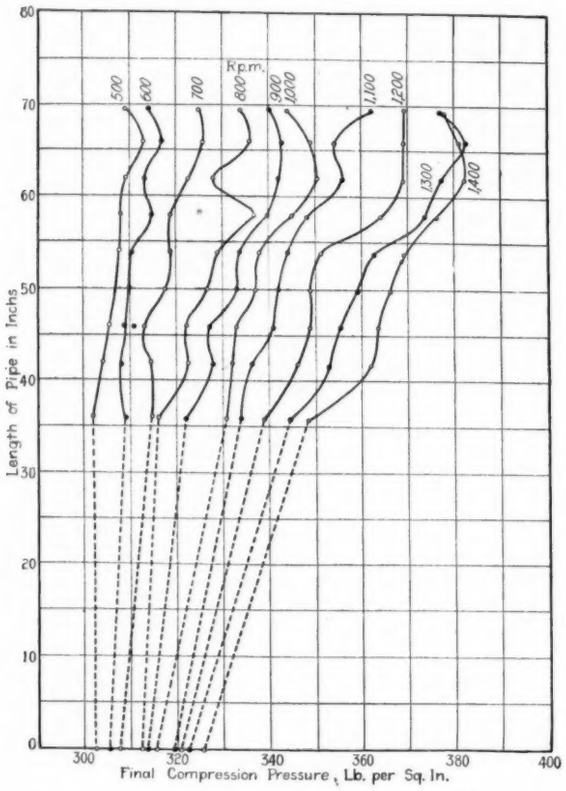
A recent issue of *Oxyacetylene Tips* shows how the oxyacetylene torch was used to make radiators out of discarded boiler tubing. The tubing was cut into lengths for headers and the ends closed by welding. Radiator tubes were cut and welded into proper openings in the headers. Inlet and outlet nipples were then welded at diagonal corners. When cleaned and painted these radiators were good looking and satisfactory. The cost came to about $7 per radiator. Wall coils of the return-bend type were likewise made from old boiler tubing without the use of fittings.

Increasing the Compression Pressure in an Engine by Using a Long Intake Pipe*

By Robertson Matthews and Arthur W. Gardiner

During some tests of a one-cylinder solid injection, compression-ignition engine, using gas oil, it was found necessary to increase either the jacket-water temperature or the compression pressure in order to start the engine. Experiments proved that sufficient increase in compression pressure could be obtained simply by attaching a long pipe to the inlet flange of the cylinder.

Data obtained during this investigation were made the basis of the curves in the illustration. Although



Variation of compression pressure with change of air pipe length

these data are not strictly applicable to another engine, they should give indications of what might be expected with such a set-up on an engine operating at the speeds here covered.

The engine used was a single-cylinder Liberty, 5-in. bore and 7-in. stroke, having standard cams, valves and valve timing and operating on the four-stroke cycle. The exhaust valve closed and the inlet valve opened at 10 deg. past top dead center. The inlet valve remained open until 45 deg. past bottom dead center. The piston was special, giving a volumetric compression ratio of 11.4 to 1. The long inlet pipe was made in two sections. The section nearer the engine was about 2¼-in. inside diameter and was bolted to the inlet flange of the cylinder, a gasket making this connection practically airtight. The other section of pipe was about 2¼ in. inside diameter and telescoped into the section attached to engine, with the joint between sections taped to prevent air leakage.

The pipe was open to the atmosphere with no obstruction such as a carburetor or butterfly valve. The only obstruction to direct flow of air into the cylinder, apart from the natural restriction of the valve port, was the almost right-angled turn, inherent in the cylinder design, above the valve seat.

During the test the engine was motored over with the dynamometer and especial care was exercised to maintain a constant speed while a pressure reading was being determined. The readings were made with an Okill pressure gage, and, since the air temperature at the end of compression approached 650 deg. F., the body of the gage, containing the piston and spring, was kept comparatively cool by means of wet cloths. The water jacket was kept at about 100 deg. F. for all the runs, which condition favored the maintenance of a uniform oil seal between the piston and cylinder.

The constant r.p.m. curves in the illustration show the relation between the length of intake pipe and the final compression pressure. The curves for these figures are drawn through the data points to emphasize small variations in the compression pressure that might actually exist, but which would be lost in a faired curve. The points widely separated from the general trend of the curve were checked, which fact seemed to be a sufficient reason for incorporating one set of curves drawn through the data points.

It will be noticed that in one case, at 1,400 r.p.m., with a 62-in. length of pipe, the compression pressure is 382 lb. per sq.in., or about 17 per cent greater than the compression pressure at the same speed when no pipe was used.

It might be well to emphasize that the volumetric compression ratio remained the same during these tests. Hence, there was no change in temperature such as would occur with a change in volumetric compression ratio. The increase in compression pressure was due, primarily, to an increase in the initial pressure.

*Extracted from Technical Notes 180, National Advisory Committee for Aëronautics.

Estimating the Air Needed from the Heat Value

In general the theoretical air required to burn coal goes up in proportion to the heat value. For burning straight carbon the heat value is 14,600 B.t.u. and the air required is 11.52 lb. This gives 14,600 ÷ 11.52 = about 1,270 B.t.u. for every pound of theoretical air. The corresponding figure for the available hydrogen is 62,100 ÷ 34.56 = about 1,800 B.t.u.

Since the combustible matter of coal consists chiefly of carbon with usually less than 6 per cent of hydrogen and a negligible amount of sulphur, the amount of air theoretically required can be closely approximated by dividing the heat value by 1,300. This short cut will prove very convenient for rough and ready computations. Suppose a 12,500 B.t.u. coal is to be burned with 100 per cent excess air, then the theoretical air required is approximately 12,500 ÷ 1,300 = 9.6 lb., and the actual air is 9.6 + 9.6 = 19.2 lb. If the coal contains 20 per cent of ash, the total weight of combustion products will be 0.8 lb. (from the coal) plus 19.2 lb. (from the air) = 20.0 lb. total. Of course this method should not be used where extremely accurate results are necessary.

Raw Material Awaits Utilization

ANOTHER angle to the problem of ash disposal has arisen from the use of pulverized fuel. The residue, in this case, is a fine dust practically free from carbon and in a more or less calcined state. While the amount of this material available at the present time is comparatively small, it probably will be considerably increased as a result of some of the conditions governing fuel consumption.

The disposal of ashes is often an acute problem not only as an expense, but as a limit to plant capacity. The utilization of such products opens a field for industrial venture. Money spent in commercial research may reduce the cost of disposal advantageously. One noteworthy result so far, has been the manufacture of special building materials from ashes.

The residue from powdered fuel presents opportunities for industrial utilization somewhat different from that of the familiar form of ashes or clinkers. In this instance we have a more refined product, which is in a condition ordinarily attained by drying and pulverizing. It appears, therefore, to be more easily susceptible to industrial processes.

This dust has been used as a substitute for sand. In some instances it has abrasive qualities that have been found worth exploitation. The qualities of this ash naturally vary in accordance with the kind of fuel used and other circumstances. It may resemble cement, and also the chemical constituents, such as iron, are likely in some instances to be present in sufficient quantity to make their removal profitable. Some investigations have already gone forward in regard to utilization of this product.

In view of the fact that large quantities may soon be available representing a comparatively uniform and finished material, it would appear that the subject of industrial application merits serious attention.

Pride and Education

MISPLACED pride is the greatest barrier separating many an operating engineer from the education he so greatly needs.

No educator worthy of the name recommends the "sponge" process of education, where the student merely soaks up facts. Practice—trial and error—are essential parts of a real education in any line. Take the cabinetmaker, for example. At the start of his apprenticeship he may have read in a book how to handle a mallet and chisel or he may have watched some journeyman demonstrate their use, but he was never worth two cents as a cabinetmaker until he had tried his own hands at it many times. This he generally did under the watchful eye of a skilled workman. A few clumsy strokes were made. Suggestions were given and eagerly received. Another try was made with better success. Further improvements were suggested, and little by little the budding craftsman added to his skill until his hand, eye and judgment became a perfect team. If a foolish pride had restricted the apprentice to the stealthy perusal of manuals of woodworking and to looking wise while his instructors demonstrated the use of the tools, he never would have traveled far.

Yet a condition much like this exists today among operating engineers with reference to book knowledge. They have grown up in the plant and have become competent in the every-day handling and adjustment of ordinary power-plant equipment. Meanwhile power-plant engineering has grown from a skilled trade to a highly technical profession requiring a considerable knowledge of practical mathematics, the physical sciences, economics and business methods. The engineer who is solely "practical"—using the word in its old sense—finds himself slipping back into the poorly paid rear guard. He begins to realize that his only salvation lies in rounding out his practical experience with some usable knowledge of the scientific and business principles underlying his work. He resolves to study and even squares his shoulders to the task, but false pride often tricks him out of his reward. He is not willing, like the apprentice, to exhibit his first poorly directed strokes to those who could guide and help him. He wants to blossom out completely "educated" without being seen taking the necessary intermediate steps.

It is astounding to find how many engineers are unwilling to bare their deficiencies to the light of day, even when that is the only way to mend them. Organizations of practical engineers are full of men seeking for knowledge by some mysterious underground route. Meanwhile the good road above ground is bare of travelers except for the stalwart few who have decided to march to their destination by the most direct and obvious route even if their neighbors do see them trundling the wheelbarrow.

The first step in getting an education is to admit openly that one needs it. The second is to be willing to be shown and even to be *shown up*, if that is the price of knowledge.

Safeguarding Power Contracts

LACK of thoroughness in considering power contracts costs some people a lot of money. Economic conditions frequently warrant the purchase of power on an interconnection agreement between an established generating plant and an outside system of supply, leading in some cases to increased use of purchased power, in others to the exclusive use of it, and in still others to the decision to restore private plant production as time goes on. Each case must be worked out upon its own merits, but a wise solution of the problem will always take into account the possibility of being obliged to

renew contracts upon a basis none too favorable to the purchaser.

No attack upon central station policy is here intended. It is only human to desire to make a profitable bargain in a power supply deal, and we are glad to testify that innumerable utilities supplying power to established plants and users strive to write contracts on terms that will be mutually beneficial to themselves and to their customers. Moreover, in many states the rates are very definitely fixed by public-service commissions. Long-term power contracts, however, need to be studied with vision by the man whose plant and establishment are going to be tied up with an outside supply system for years, inasmuch as outside conditions may change such as to warrant increased rates and his own operating conditions may become more favorable to private generation. Local generating plant machinery often stands in reserve in such cases for long periods, sometimes deteriorating, falling into obsolescence and perhaps being removed to make way for process machinery or for storage of products or raw material. Sooner or later the day comes when the question of renewal or further supply must be faced, and in some instances the purchaser finds himself "up against" a serious problem, namely, how to renew at a profitable rate to himself in view of existing conditions or how to re-establish his own generating facilities at reasonable cost.

It is fruitless to expect that power prices can be guaranteed to continue unchanged after the expiration of contracts; but a good deal can be accomplished toward insuring a mutually satisfactory arrangement or an economical return to local generation by exercising a little more vision in the first place. As long as local generating plant equipment is held in reserve, it pays to keep it in good shape for service. An alternative source of supply is a potent argument in negotiating for the renewal of a contract in sight of expiration. Then, too, provision for arbitration in case of disputes and a carefully planned procedure relative to renewals or extensions, saves a great deal of trouble.

Court and commission authorities have held that just because a power contract proves unprofitable, it cannot be set aside unless it is distinctly in the public interest to abrogate such an agreement. It is far better to outline a course to follow in case a contract does prove unprofitable than for either party to be gripped as in a legal vise; unable to secure a fair readjustment and forced, perhaps at great loss, to continue an undesirable arrangement long after it is economically outlawed. By applying a little imagination and good sense to future possibilities, much grief can be avoided and advantageous contractural arrangements maintained by all concerned.

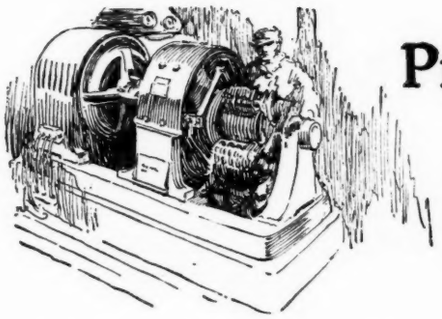
Another Opportunity for Co-operation

WHERE high-head installations are encountered, the penstocks and pipe lines become a real factor in the reliability of a water-power installation, as is attested by a number of failures of these parts. Failure in a pipe line or penstock can occur from two general causes—excessive pressure set up by arresting the flow, and a reduction of pressure below atmosphere, which would result in a collapse of the water passage. In the low- and medium-head plant the governing time of the turbine can be practically ignored in so far as the penstocks are concerned. In high-head plants the time in which the governor can open and close the turbine gates is an important factor and one that must be given proper consideration in the design of the penstock.

Contrary to what might be expected, the time of opening the gates may be of greater importance than the closing time. By a properly designed relief valve, made to operate in synchronism with the turbine gates, excessive rises in pressure can be avoided in the penstock when the turbine gates are closed. When opening the gates, if the time is made short, a bad surge may be set up in the penstock, especially if there are flat gradients of considerable length at the top end of the penstock with a steep gradient dropping down to the turbine. If the gates are suddenly opened, the water will be accelerated much more rapidly at the lower end and in the steep gradient than at the upper end of the penstock, with the result that the water column tends to part, resulting first in a reduction in pressure and then in a rise when the water in the upper half of the column has been accelerated to where it has caught up to the lower part. This surge is the most difficult to provide protection against and, in fact, can be controlled only by the governing time after the penstock has been built. By proper design much can be done to avoid layouts that tend to produce undesirable flow conditions.

Another feature in the design of high-head penstocks is what is to be considered normal working pressure. Is this to be the static head of the water, or is some allowance to be made for the rises in pressure due to uncontrollable surges? A penstock in its operation is different from most pressure vessels in that it is expected to be subjected to pressures considerably above the normal working value. In most pressure vessels the pressure rise is at a comparatively slow rate, and by the use of properly designed safety valves the stresses can be maintained within normal working pressure. A penstock may not only be subjected to sudden rises of pressure above normal, but also to decrease below normal, and these may occur in different parts of the penstock at different times, depending upon how these changes are set up. Furthermore, these changes may be of such a nature that it is impossible to provide protection against them. These, what might be called normal changes in pressure, are dependent upon the time of governing, gradients in the penstock static head, etc., and it would seem that the most logical conclusion would be to consider these as normal and make the design for these pressures. If operating conditions are such that a one hundred per cent rise above normal static pressure may occur, then the penstock should be designed for double normal static head and the specified factors of safety based on this pressure. The conditions that may exist in the penstock after its installation are so dependent upon the turbine's operation, as to justify very close co-operation between those who design the penstocks and the turbine builders. Although the high-head penstock failures that have occurred during the last two or three years may not give cause for alarm, they do show that this problem merits serious consideration by everyone concerned with hydro-electric developments.

A reader submits a sample package of another coal saver. This one is put out by the Wiley Products, Inc., of Long Island City. The usual directions for its use are prescribed and like all the rest the claim is made that it "Saves ¼ to ½ your coal." The spirit that Barnum exploited is still apparently prevalent.



Practical Ideas *from* Practical Men

Why Not a Gas-Steam Binary System?

The desire for higher efficiencies in power plants has led to an experimental binary system where two substances, water and mercury, are used. The cost of such a system is high, and it is surprising that investigators have not turned to the combination of steam and producer gas, which will not only give a far greater efficiency than the proposed binary system, but should be much cheaper to install. In addition the byproducts resulting from the producer distillation would be a source of a considerable income.

Some years ago the Mond producer was installed in a large way. This system of producing power gas from inferior soft coal involved the use of large quantities of steam. Having this in mind, it occurred to the writer that an economical power plant could be developed through the medium of a system employing gas producers of the Mond or similar type.

For efficient ammonia production this producer uses about two tons of steam per ton of coal gasified. The gas from such producers could be burned at a pressure of, say, 150 lb. (air at the same pressure) in a steam boiler designed to abstract enough heat from the products of the combustion to permit these hot gases at 150 lb. to be fed directly into a suitable gas turbine. Now the amount of heat necessarily removed from the products of combustion by this special boiler, may be only sufficient to provide the steam required by the producer. But any steam in excess of that amount could be used to drive a steam turbine. And the steam required by the producer could first be utilized in a unaflow engine, which operates well under the moderate back pressures needed for the producer, to make use of its available power. Thus the heat abstracted in cooling the gases for use in the gas turbine, need not be wasted and the latent heat of the steam used in the producer is largely conserved.

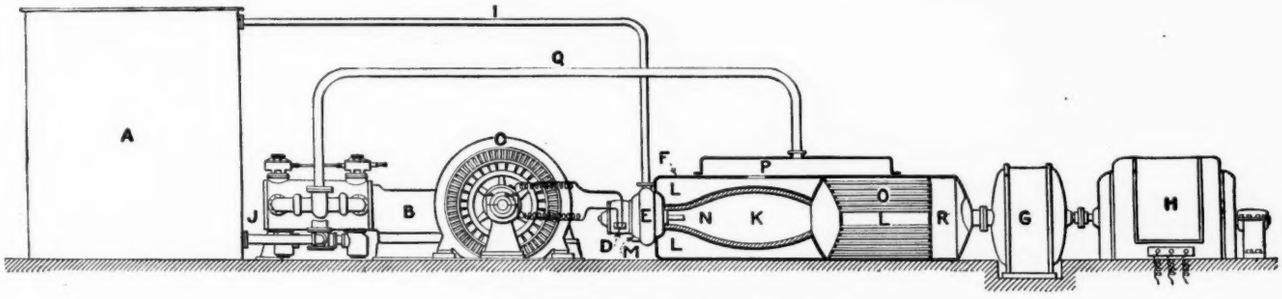
With regard to the temperatures of the exit gases from the special boiler or cooler, these gases could be conducted directly to the main power turbine and so provide the chief motive fluid. These gases could range from 1,200 to 1,500 deg. F. or a temperature that experiment shall determine to be feasible for a properly designed turbine. By utilizing recently developed alloys containing chromium and nickel, which retain most of their strength even at low red heat, no real difficulty should be experienced in developing a suitable turbine.

It would seem as though a power plant along the lines shown diagrammatically in the illustration, is within the scope of modern engineering knowledge and might offer important heat economies over the orthodox plant.

Then again a simpler arrangement is conceivable in the shape of a combination power plant of steam turbines and turbines operating on fuel-oil gas. The oil would be burned, say, in the type of boiler previously suggested. Again, this boiler would be designed to abstract just the heat necessary to reduce the temperature of the gases of combustion to a limit where they may be safely introduced into the gas turbine.

The weighty doubt in all these schemes is the compressing of gas and air. I have read of an experimental, single-stage, high-pressure rotary air compressor having an efficiency of 64 per cent. Such a compressor could be built into the boiler and form an integral part of it, so that it could be cooled by the boiler water with but slight heat loss.

May not the basic defect in large power production be due to the use of an external medium—steam—as motive fluid, instead of the employment of the products of combustion of the fuel? This entails two great heat losses: (1) Most of the latent heat of the steam; (2) the sensible heat of the flue gases. Yet if the Diesel engine be adopted in an attempt to obviate such losses, there is encountered: (1) Limitations of size, etc., due to reciprocating action and high heat; (2)



Suggested Gas-Steam Plant

A, Mond producer; *B*, unaflow engine; *C*, electric generator; *D*, motor driving gas blower; *E*, gas blower; *F*, steam boiler; *G*, gas turbine; *H*, electric generator; *I*, gas line; *K*, combustion chamber; *L*, water jacket; *M*, air inlet to blower; *N*, gas-air nozzle; *O*, boiler tubes; *P*, steam space; *Q*, steam line; *R*, flue-gas chamber.

heat loss in jacket water. It would seem as if some of these disadvantages could be overcome through the system outlined.

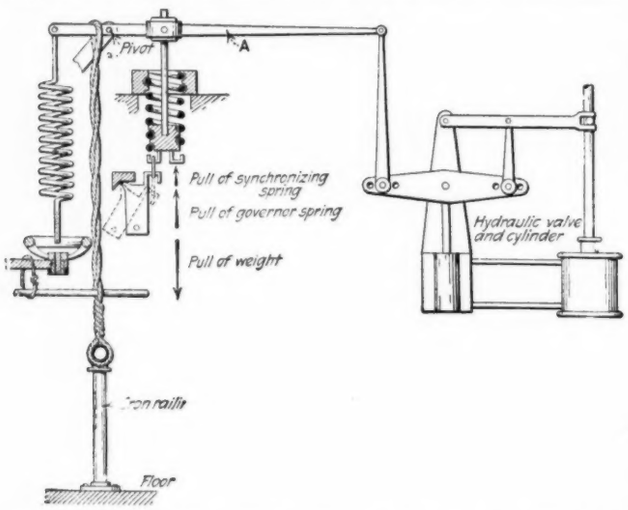
While some of these suggestions may be a bit visionary, is there not room for a little unorthodox thinking on an industry which operates, at best, at not over 20 per cent efficiency? ALFRED COCHRAN.

Brooklyn, N. Y.

Operating Without a Synchronizing Spring

The figure illustrates a quick means of applying tension in case of the sudden failure of a synchronizing spring. In this case an assisting type spring stripped its holding threads while a large turbine was carrying full load. The generator lost its load and operated as a motor drawing current from the busbars. Instead of shutting down the unit for repairs, the main switch was kept closed so that it continued in synchronism with the other units.

A 1-in. rope sling was obtained and used as a tourniquet for applying tension, as illustrated. The iron



Turbine maintained under load in spite of synchronizing spring failure

railing which held the lower end was firmly attached to the flooring. By twisting up the rope, it was easy to apply tension until the machine picked up the desired amount of load. Flexibility naturally present in manila rope made it a good substitute for a spring. The synchronizing apparatus was repaired, while operating, and a few hours later normal conditions were restored.

Synchronizing springs in general are of two characters. The first is one which assists the governor spring as in the figure, and the second is an opposing type which works against the governor spring. It will be noticed in the assisting type illustrated, that the pull of the synchronizing spring, through the governor lever, acts *with* the pull of the governor spring. The arrows will show how these two work together and oppose the pull of the weight. Tightening such a spring will therefore tend to speed up the unit, and increase the load. If the spring lets go, the unit will drop in speed, and if carrying an alternating-current load, it will quickly lose the load and be run as a motor by the current from other units with which it is in parallel.

With an assisting-type spring the machine in all probability could not be run at full speed. It would not then be possible to put this on a line with a synchronizing spring out of commission. One way to operate would be to disconnect the governing mechanism, block the valves open and operate from the throttle entirely. Another way would be to substitute another means of tension for that of the synchronizing spring.

If tension is applied, however, by means of hanging a weight on the governor lever, the latter method will, in all probability cause a hunting or surging of the governor and valve gear. This introduces a large mass into the mechanism which will tend to make the operation of the valve gears somewhat slower since this mass must be moved each time the governor weights change position. A means of applying tension without adding to the weight perceptibly is, therefore, to be desired.

In the case of an opposing spring the illustration would represent a spring in compression instead of tension. In this case the turbine could be governed directly from the throttle, since it would naturally be much above speed without the tension normally supplied by the synchronizing spring.

Closing down on the throttle would therefore take care of the load conditions while repairs were being made. However, tension could be applied, if desired, by shifting the rope to the governor lever on the other side of the operating pivot, such as at the point *A*, provided it were strong enough to take the pull at this point. Many readers could doubtless add experiences along this line. G. F. WHITE.

New York City.

Care Necessary When Operating Voltage Regulators

An incident occurred in a large power house when the operator, in taking the regulator out of service, turned the main exciter rheostat, across which the contacts operate, toward the "all-in" position instead of toward the "all-out" position. The result was that the regulator, in an endeavor to maintain the voltage, slowed down in its vibration as the rheostat was cut in, until finally the contacts remained closed. The operator then opened the disconnecting switches to the relays, thinking the contacts were open, which would be the case if the rheostat had been turned toward the "out" position. The result was an immediate reduction in exciter voltage when the switches were opened, with the consequent loss of load. The exercising of a little caution would have prevented this case of trouble.

Inasmuch as regulators play an important part in the operation of power plants, it is advisable that the station be laid out so that the hazards are a minimum and the operation as simple as possible. In order to attain these ends, the use of individual regulators reduces the complexity of operation, and in cases of trouble with the regulator it does not affect the entire station, but only one machine. The use of individual regulators is always to be preferred.

Automatic voltage regulators like any other piece of apparatus must have intelligent handling if they are to give the service they are intended. The trouble referred to in the article, "Generator Automatic Voltage Regulators—Care and Operation," Feb. 9 issue, is only an example of what may happen either from improper operating conditions or a mistake in handling equipment.

New York City. A. A. FREDERICKS.

Comments from Readers

Where Should the Feed Water Enter the Boiler

I read with interest the set of boiler questions by Mr. Hilleary in the Jan. 22 issue. In the answer to question 26 he says feed water should enter preferably on the opposite side from the water column. I would like to ask why? I have operated Scotch boilers in land service that were fed through the bottom blowoff and gave good results. I also operated one that was fed in the front head under the furnace shells and never experienced any trouble.

Commenting on question No. 29, "Where should the feed water enter a vertical tubular boiler?" I had a small vertical dry-end tube boiler in which the feed water at 160 deg. temperature on entering the boiler was spread across the lower tube sheet. As a consequence this sheet was leaking most of the time. Upon altering the feed connection so that the water entered about six inches from the top of the tube sheet, the lower sheet stopped leaking. Changing the location of the feed inlet had no apparent effect on the steaming of the boiler. R. G. SUMMERS.

Rochester, N. Y.

How Motor Cables Were Connected Quickly in an Emergency

I should like to make a few comments on the article in the Jan. 29 issue entitled "How Motor Cables Were Connected Quickly in an Emergency."

In the plan described in this article small cones were formed out of sheet iron, their lower ends fitting closely around the wires to be tapped after the insulation on those wires had been removed. The small ends and the seams were sealed. Then the leads to the motor, their ends having been bared, were hooked over the tops of the cones and their ends allowed to hang down inside. The connection was completed by filling the cones with molten solder. When the solder had cooled, the connections were taped up and considered complete.

In these connections the leads to the motor were not necessarily touching the mains and the only path between the two sets of wires was through the solder.

Owing to the fact that the solder has not the mechanical strength of copper wire, has a relatively high electrical resistance and a low melting point, it would seem that the connection was not one to be recommended. A sudden blow might have torn the lead away from the main, or the high resistance of the solder, unless a large amount were used, might cause the joint to heat up and soften the solder, loosening the connection still more.

It was stated that the reason for making that kind of a connection was to save time, but a good wireman could have made a wrapped joint in almost the same time that was required to make this joint and then wait until the solder had cooled sufficiently to permit of current being passed through it with safety. This wrapped joint would have been as good electrically and better mechanically than the joint described and, unless the voltage used on the circuit was dangerously high, could have been soldered later while carrying current, thereby making a first-class job.

Neither the voltage on the circuit nor the size of the wire used in the leads is mentioned in the article, but the illustration accompanying it showed a stranded conductor and the size of the motor connected up, 75 hp., would probably make that type of wire necessary unless the voltage was very high. G. H. MCKELWAY.

Westfield, N. J.

Once More, the Throttling Calorimeter

In the Feb. 5 issue of *Power* Prof. E. H. Lockwood criticizes the design given by the writer for a throttling calorimeter made from standard pipe fittings, and gives a sketch of the calorimeter designed by George H. Barrus, which also has been extensively used in power-plant work.

Without doubt the comments of Professor Lockwood should be given much weight. I would, however, like to point out a few reasons why I have never regarded the Barrus design better than that presented in the Dec. 25, 1923, issue.

In my design the drop in pressure takes place through the narrow annular slit formed when the valve is very slightly raised from its seat. The jet of vapor is quickly broken by the eddies within the ramifications of the valve, where it must also turn a right angle before entering the nipple and passing to the space surrounding the thermometer well. It would be surprising if the velocity energy of the steam has not been reconverted to heat before this point is reached.

In the Barrus design the pressure drop takes place through a small round hole in a thin plate. Here the jet is a solid round stream directed along the center of the pipe and is a much harder jet to break up than the thin, sheet-like jet in the other apparatus. Eddies surrounding the solid round jet have little influence in breaking it up, and the elbow turn will scarcely complete the work. In spite of the greater distance and elbow I doubt if velocity energy is any more completely lost in the Barrus design than in my own.

Furthermore, we might bear in mind that before the velocity energy is reconverted to heat, the temperature within the jet is that of saturated steam at about atmospheric pressure, or 212 deg. F. If this steam impinges on the thermometer well, can it heat that body to a higher temperature than it can superheat itself. I may be mistaken, but I do not think so. On the contrary, I would expect the thermometer in the stronger steam jet to show the lower temperature, because the high velocity steam is cooler, and wet.

This may explain why, in Professor Lockwood's tests, the Barrus calorimeter showed a lower temperature, if my reasoning is right that in the Barrus design the velocity of steam is greater. A second possibility is

that, owing to the greater surface in his design, radiation losses may result in the lower temperatures observed. In my own design I followed the original Peabody calorimeter in reducing the radiating surface to a minimum.

What Professor Lockwood mentions as the defective type of calorimeter may have had an orifice so placed that its velocity was directed straight against the thermometer well. This is not the case in my design, as I have explained.

In an earlier paragraph I asked the question as to whether the impinging of wet steam at 212 deg. on a body can heat that body to a higher temperature than the steam can superheat itself. I ask this for information. I have conducted many tests that might have shown this phenomenon if it were possible, but have not seen it. I have been told by engineers of cases where it appeared to have happened, but none were well authenticated. It does not look to me reasonable, and therefore I would like to have proof. THOMAS M. GUNN.

New York City.

Superheated Steam Used Directly in Closed Heaters

In the Jan. 29 issue, Mr. Sprague develops a theory to prove why superheated steam used in closed water heaters has the same effect as saturated steam, the pressure and other conditions being the same. He supposes that the superheated steam is never in direct contact with the tube surface and that a film of condensed water covers the metallic wall. But generally this condition is not fulfilled and therefore in my opinion the conclusion of Mr. Sprague is wrong.

The steam follows a certain path. If originally superheated, it first loses the superheat, then is condensed and finally the condensing water is undercooled. These three zones normally do not mix. There exists, therefore, a zone where superheated steam transfers its heat through the wall of the tube to the water on the other side, with a poor effect of heat transfer.

Regarding only the rate of figures, we can suppose the coefficient of heat transfer to be for superheated steam one fiftieth that of water and for a saturated steam five times as high as for water. We will neglect the insignificant influence of size and material of the pipes. The coefficient of heat transfer for superheated steam to water is to the coefficient of transfer from saturated steam to water as

$$\frac{1}{\frac{1}{1}+\frac{1}{\frac{1}{50}}} : \frac{1}{\frac{1}{1}+\frac{1}{5}} = \frac{1}{51} : \frac{5}{6} = 1 : 42.5$$

The mean temperature difference between water and steam must therefore be for the superheated zone, about 40 times as high to arrive at the same heating capacity as when working with saturated steam. As normally, the mean temperature difference between superheated steam and water can be supposed to be not more than ten times as high as between saturated steam and water, the B.t.u. transferred per unit surface in unit time is about one-quarter using superheated steam instead of saturated steam.

Mr. Sprague states that he has made numerous heat-transfer tests proving the correctness of his theory. Of course there are exceptional cases for which the general conclusions made before are not available. If, for example, the velocity of the water is very low, (which means a wrong construction) and the velocity of steam very high, the heat transmission becomes almost independent of the quality of steam and is limited only by the poor amount of heat transfer between water and tube.

Furthermore, it is possible to use special constructive measurements to fulfill Mr. Sprague's condition of covering the total surface with a film of condensed water, for example mixing the entering superheated steam with the undercooled condensing water. This means, of course, a special manner of desuperheating the steam and results obtained with such constructions cannot prove the equality of heating effect of superheated and saturated steam. M. HIRSCH.

Frankfurt, Germany.

Chasing Bubbles in a Boiler

The article in the Feb. 19 issue entitled "Chasing Bubbles in a Boiler," is interesting, but I wonder if the circulation as shown in the glass model is exactly what happens in a real boiler. It appears doubtful to me. The only way in which to be absolutely certain about the circulation is to make a glass boiler of full size. This has been demonstrated, for example, in connection with steel and concrete beams and columns. Small models of beams cannot be used for carefully computing the strength of the full-sized members. The stresses in models and large members are not the same. In the same way it is possible that the action within the model boiler is not the same as that within a large boiler, because of the difference of capillary attraction. In the small model capillary attraction certainly plays a more serious or important rôle than in a large boiler. The tubes are doubtless much closer together than in a large boiler and therefore the greater resistance to circulation very likely influences the path of the water. In a full-sized boiler the circulation of the water therefore may be exactly as it has been supposed to be, as shown in Fig. 1 of the article referred to.

One of the most interesting exhibits at the last power show was the one by the Bailey Meter Co. in which full-sized pipes made of glass, demonstrated the action of water in flowing through pipes. That demonstration was an eye opener to me. Some of the other exhibits, however, particularly some of the models, did not arouse much enthusiasm in me because of my knowledge of the behavior of models in general as compared with full-sized equipment, and simply because a tiny boiler does so and so is no proof to me that a full-sized boiler will do the same thing. L. W. LOWERRE.

Elizabeth, N. J.

Heat-Work Conversion Law Questioned

I would like to place before the readers of *Power* a simple demonstration of the fallacy of a commonly accepted thermodynamic "law." During the last four years this demonstration has been submitted to a large number of scientists, professors and practicing engineers. Of the many answers submitted none has, in my opinion, been direct and satisfactory. If there is any fallacy in my demonstration, it ought to be possible to point it out in a simple and direct manner without equivocation.

The figure shows a perfect gas-engine diagram (Otto cycle) in which the curves *HC, IN* (not shown) and *JL* are adiabatic. For ease of illustration the thermal

efficiency or heat conversion has been fixed at 50 per cent. In the cycle *LJHCL*, 100 B.t.u. is supplied along *JH*, 50 B.t.u. is rejected along *CL*, and the remaining 50 B.t.u. vanishes. These even numbers have been chosen to simplify the presentation. The conventional idea is that the useful work *LJHCL* is received as full equivalent of the vanished 50 B.t.u. Such idea has not been rejected, as one might suppose, but has been rigidly accepted and consistently adhered to; that is, for present purposes, of course.

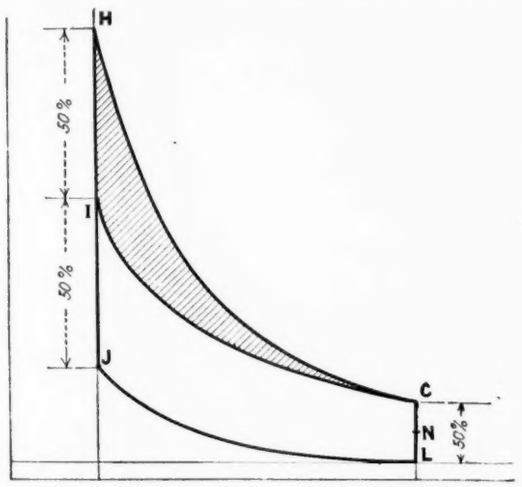
For the first time, apparently, the conversion concept has been taken at face value, resolved to its lowest terms and then visualized by graphically setting forth the presumed facts in the simplest of diagrams. In any heat engine in which an elastic working medium is manipulated, the heat involved at the start of the working stroke comprises a definite portion which shall conventionally be converted, and another portion which shall likewise remain unconverted. In order accurately to accredit to each of said portions the actual work done by each during the expansion stroke, the so-called "phantom curve" *IC* is plotted so as to be at all points just 50 B.t.u. above the line *JL*. The shaded area *IHCI* thus represents the *sum* of the work done by converted heat, while the unshaded area *LJICL* must naturally represent the *sum* of that done by the non-converted.

This can be proved as follows: In the engine cylinder the "converted" 50 B.t.u. of heat of combustion is *of* the confined volume at start of working stroke and is *not of* the same at the end of such stroke. At the start, its presence acts to maintain some portion of the excess pressure (above compression pressure), and at the close, being non-existent as heat, it of course maintains none of the pressure at that point. Therefore, since pressure only will do work against a piston, and as this pressure is due solely to the actual presence of heat, as heat, it is clear that the actual physical work done by this "disappearing" portion is definitely portrayed by the process set forth. We may now ask question 1: If the converted portion is competent to do more than the specified work (shaded area *IHCI*), then how, when and where is such additional work done?

Assuming that the foregoing is strictly in accord with the presumed facts, let us proceed to the point next in order. Now, if but a half portion of heat (50 B.t.u.) be supplied along *JI* for the next working stroke, the pressure during expansion will naturally fall on the adiabatic *IN*, and the sum of the useful work done by such heat will be accurately represented by area *LJINL*. Then question 2 arises: How was an identical half-portion enabled "supernaturally" to perform considerably more (area *ICNI*) than this normal quantity of work in the previous instance, when it was merely incidentally associated with a like quantity which gradually vanished from the volume confined within the engine cylinder?

If the term "approximately adiabatic" be substituted for word "adiabatic," and if the pressure increment *CL* be defined as representing "50 B.t.u. less radiation losses," it will be clear that the foregoing illustration involves but one debatable assumption, that of heat-work conversion. Confronted with the impossibility of answering this second simple and perfectly obvious question in a way to satisfy even an amateur, the engineer naturally turns to the recognized proofs; the "standard" proof, of course, being Joule's experiment in which water is heated by stirring with a paddle. To limit the discussion to the case in hand, all such proofs are conceded to be conclusive—for such types of "engines."

Commercially, all so-called heat engines in power plants are today employing an elastic working medium in the engine or turbine which serves as a "heat-work converting" means. In such types of engine (using the reversed gas-engine process as an example), work energy may conventionally be "mathematically" converted into heat; that is, provided the precaution be taken first to arrange for a continuous supply of physical heat to the engine. Unfortunately, however, and in spite of any and all figures to the contrary, the total heat actually delivered will be precisely that quantity which has been previously supplied as heat—no more, and no less. Cut off from such an engine its heat



WHAT WORK DOES THE HEAT DO?

supply without disturbing the work-energy supply, and delivery of either physical or "mathematical" heat will cease forthwith. ("Mathematical" heat, so-called, is simply temperature, or what may be termed "heat-pressure," the same being erroneously accepted as physical heat by the profession and mathematically "proved" to be such.) If these latter assertions be untrue, it should not be a difficult matter to set forth a cycle layout, to cite a concrete example, or to stage an actual physical test which shall effectually dispose of them.

Should the readers of *Power* seriously contemplate the fact that the modern internal-combustion engine conforms very nicely to present thermal-efficiency formulas, and also the extraordinary fact that such engine actually produces hot water as its greatest product, it might occur to them that all may not be well with certain accepted practice, and they may therefore confidently expect that the issue in hand will be followed through to a definite conclusion.

New York City. WILLIAM R. WILLIAMS.

[Although Mr. Williams' views in this matter are in direct contradiction to those of practically all scientists and engineers, *Power* is glad to allow him this opportunity to state his proposition. Space restrictions will make it necessary to limit the discussion on this proposition to a few of the letters that seem to answer Mr. Williams' contention most clearly and concisely.—Editor.]

QUESTIONS AND ANSWERS

Conducted by Franklin VanWinkle

Required Size of Boiler Grates

What would be the size of grates under a boiler to evaporate 2,359 lb. of water per hour, not burning over 15 lb. of coal per square foot of grate per hour and an evaporative power of 8 lb. of water per hour per pound of coal? *E. B. F.*

The coal consumption would be 2,359 ÷ 8 = 294.8 lb. per hour, and burning 15 lb. of coal per square foot of grate would require 294.8 ÷ 15 = 19.6, or practically 20 sq.ft. of grate surface.

Relative Viscosities of Water, Kerosene and Gasoline

What are the relative viscosities of water, kerosene and gasoline at the same temperature? *S. H.*

The viscosity of gasoline at atmospheric pressure is taken at 62 deg. F. in order to be below the boiling point. Tested with a Saybolt viscosimeter with the temperature in each case 62 deg. F., the viscosity of water at 62 deg. F. is 30; of kerosene, 40 to 50, depending on composition; and of gasoline, 35 to 45, depending on composition.

Slagging Between Tubes of Water-Tube Boilers

We have two water-tube boilers equipped with chain-grate stokers, flat arches and soot-blower units. With a few days' hard service of a boiler the spaces between the tubes at the front end, up to the third row, become completely choked with slag, and the blower unit near the fire becomes burned out. How can these troubles be prevented? *G. F. B.*

Undoubtedly, the choking up of spaces between the tubes results from infrequent use of the soot blowers or their inefficient operation from becoming burned out. To prevent burning out of the soot blower unit, it should be protected from direct radiation of heat from the fire by some form of baffle.

Reduced Thickness of Boiler Plates at Circumferential Joints

Why are thick boiler plates of horizontal return-tubular boilers reduced in thickness at circumferential joints? *J. P.*

When the shell plates exceed $\frac{9}{16}$ in. in thickness, the portions of the plates that are lapped to form a circumferential joint, where exposed to the fire or products of combustion, should be planed or milled down to $\frac{1}{2}$ in. in thickness, so the plates may be thin enough for plates and rivets at the joint to be kept cool enough by the water of the boiler to prevent their becoming burned. It is admissible for the strength of joint to be thus reduced to only 35 per cent of the strength required for the longitudinal joints when 50 per cent or more of the load which would act on an unstayed solid head of the same diameter as the shell, is relieved by the presence of tubes or through stays, and the holding power of tubes and stays.

Earlier Compression Required When Operating Condensing

After the season is over for heating our building with exhaust steam, our four-valve engine will be hooked up with a condenser. What changes should be made in the valve setting? *W. B. J.*

In operating condensing, the lower pressure of the exhaust will require the exhaust valves to be closed earlier to obtain the same cushioning effect by compression as when operating non-condensing. In other respects the requirements of valve setting would be the same whether operating condensing or non-condensing.

Loss from Leaving Vertical Boiler Uncovered

We have a vertical fire-tube boiler 48 in. in diameter by 8 ft. 4 in. high, operated at an average pressure of 90 lb. gage. This boiler is not insulated. What is our average loss by direct radiation? The cost of coal used is $11.50 per ton. *G. R. C.*

The loss of heat by radiation from the boiler shell would be about 3 B.t.u. per degree difference of temperature inside and outside, per square foot of surface, per hour. The temperature of steam at the pressure of 90 lb. gage is 331.4 deg. F. Assuming the average temperature of air surrounding the boiler to be 70 deg. F., the average temperature difference would be 331.4 — 70 = 261.4 deg. F. Deducting firedoor and handhole openings, the area of boiler shell that should be covered would be about 103 sq.ft., and under the stated conditions, the loss of heat would be 3 × 261.4 × 103 = 80,773 B.t.u. per hour. A good quality of non-conducting covering 2½ to 3 in. thick would have an efficiency of at least 80 per cent and prevent the loss of 80 per cent of 80,773 = 64,618 B.t.u. per hour.

The amount of coal required per hour to make up this loss of heat would depend on the evaporative economy of the boiler. Allowing that this is equivalent to an average evaporation of 6½ lb. of water from and at 212 deg. F. per pound of coal, the heat obtained in steam per pound of coal used would be 6½ × 970.4 = 6,308 B.t.u. Hence, to make up loss of 64,618 B.t.u. per hour would require the consumption of 64,618 ÷ 6,308 = 10.24 lb. of coal per hour. Using the boiler 8 hours per day for 300 days per year with coal costing $11.50 per ton of 2,000 lb., the loss that could be saved from the use of a good insulating covering for the boiler would amount to 11.24 × 8 × 300 × $11.50 ÷ 2,000 = $155.11 per year.

Equality of Cutoff of Corliss Engine

After setting the valves of our Corliss engine to obtain equal cutoffs in opposite ends of the cylinder, with little more than friction load on the engine, we are disappointed to find there is considerable inequality in the diagrams from the engine when fully loaded. What should be done to correct this difference? J. J. Y.

With ordinary Corliss valve gearing the same relative cutoffs are not obtainable throughout the range of cutoff. For that purpose the design would have to be symmetrical and the adjustment, friction and wear of latches and other connections to each steam valve would have to be idéntical in all respects. Hence valve setting that gives equality of cutoffs with a light load cannot be expected to give the same relative cutoff for other loads. The best that can be done is to adjust the governor reach rods to obtain equality of cutoffs for the average load. However, if the engine is occasionally called upon for development of its greatest power, the adjustments should be made with reference to obtaining the latest cutoff from both ends. Although this may result in considerable inequality of cutoffs with the average load, it will not materially affect the economy so long as a good indicator diagram is obtained from each end of the cylinder.

Method of Packing Pistons of Hydraulic Elevators

Describe the process of packing the pistons of hydraulic elevators. W. H.

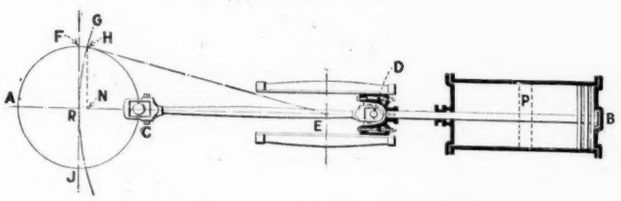
On horizontal-type cylinders the car must be taken to the highest point of travel and secured by means of a chain block and also by setting the safeties. The stop valves are then closed, the water drained from the system and the packing gland removed. The piston is now at the extreme outward travel; the strain is off the hoisting cables; the follower or gland is loosened and removed from the piston. The old packing is carefully cleaned away and new packing inserted. The new packing generally is one-inch square, and on a 26-in. diameter cylinder it is proper to cut the packing from two to four inches longer than the internal circumference of the cylinder and force the packing in by means of fiber tools, it being always necessary to butt the ends together and insert them first. Joints should always be broken. After the follower has been firmly and evenly tightened up, each nut should be loosened two threads to allow for expansion.

In the vertical-cylinder type, if conditions permit, packing should be done from the bottom, the operation being much less difficult than packing from the top. When the piston is packed from the bottom, the car is taken to the highest point of travel and is secured in the same manner as described for the horizontal-cylinder types. The stop valves are closed, and the water is drained from the cylinder and valve chambers. The bottom piston head is removed and access is then had to the piston. The follower is taken off and the cup leathers (if any) and the packing removed. The same cup leather may again be used in many cases, in which event, after all parts of the piston and the cylinder walls below the piston are thoroughly cleaned, the cup leather is set back in position and three or more rings of square packing inserted, with joints broken. After the three or four rings of packing have been forced into position, the follower or gland is returned to its position on the studs and made quite tight. The follower and piston head are then replaced and the equipment is ready to be filled with water. Before releasing the car on any hydraulic, after repairs to the valves or packing have been made, it is of the utmost importance that all air in the system (except that in the pressure tank) should be excluded and that full pressure of water be on all parts of the system.

Requirement of Unequal Lap for Equal Cutoff

For obtaining cutoff at the same fraction of stroke from the head end as from the crank end of a D slide-valve engine, why is it necessary for the valve to have more lap on the head end than on the crank end? M. C.

The motion of the valve depends on the angular motion of the eccentric, which must be the same as the angular motion of the crank. However, on account of the angularity of the connecting rod, fewer degrees of rotation are accomplished for any fraction of stroke from the head end than for the same fraction of stroke from the crank end. Referring to the figure in which *AB* is the center line through the shaft and cylinder, *CD* the length of connecting rod and *CJAFH* the circle described by the center of crankpin *C*, it may be



Angularity of Connecting Rod

observed that when the piston is at *P*, halfway through a stroke from the head-end dead center, with the crosshead end of the connecting rod carried to *E*, the connecting rod *CD* would be long enough to reach from *E* to *R*, the center of the crank circle, but could not reach to a point *F* because, in the right-angle triangle *ERF*, the hypothenuse *EF* must be greater than the base *RE*.

If, with *E* as a center and radius *ER* = *CD*, we describe the arc *RG*, it intersects the crank circle at *H*, showing that half-stroke from the head end is accompanied by less than 90 deg. rotation of the crankpin, and the distance *NC* from the foot of a perpendicular dropped from *H* upon *AB* would be equal to the horizontal movement of the crank for one-half stroke from the head end, and *AN* would be the horizontal movement of the crankpin for one-half stroke from the crank end.

In the same way it can be shown that for any fraction of stroke from the head end, there is less angular movement of the crank and crankshaft than for the same fraction of stroke from the crank end. Hence, with equal laps on the valve the cutoff would occur later in the stroke from the head end and the discrepancy increases the shorter the length of connecting rod in proportion to the length of crank. This discrepancy of cutoffs cannot be remedied by moving the eccentric, since setting the eccentric ahead will make all valve events earlier in both strokes, or turning it backward will have the reverse effect. The cutoff can be equalized by adding to the head-end lap and deducting a similar amount from the crank-end lap.

[Correspondents sending us inquiries, should sign their communications with full names and post office addresses.—Editor.]

Here and There in the Power Plant

Sidelights on things generally used but less generally understood

How Water Power Is Figured

THE power that may be obtained from any hydraulic installation depends on three things—the volume rate of flow, the available fall and the efficiency of the plant. From one point of view this question of the power obtainable from flowing water is very simple. From another it is highly complex. It is easy enough to figure the power available under conditions as they exist at any given moment, but it is difficult to decide upon the proper size of plant. The latter involves a thousand and one factors and cannot be properly handled without making an extensive study of construction costs, size and nature of market for energy, rainfall variations throughout the year, extent of watershed, the possibility and cost of various kinds of storage, the use of steam plants, etc.

What Is Work?

The discussion of these problems is far beyond the scope of this article, whose only purpose is to show how to answer the simple mechanical question of the power available under given conditions. This will be done in terms of the fundamentals of mechanics: Distance, time and force, expressed in feet, seconds and pounds respectively.

The first step is to combine distance and force to get *work*. If any body moves while under the action of a force, work is done, the amount of this work in foot-pounds being the product of the force in pounds by the distance in feet which the point of application of the force moves *in the direction of the force*. This last qualification is important. For example, if a weight of 60,000 lb. is lifted vertically 10 ft. the work done is 60,000 × 10 = 600,000 ft.-lb. If, however, a 60,000-lb. freight car is hauled 10 ft. up an incline, the work done is much less than 600,000 ft.-lb. If the bearings are frictionless and the vertical rise in 10 ft. of track, is, say, $\frac{1}{10}$ ft., the actual work done on the car is 60,000 × $\frac{1}{10}$ = 6,000 ft.-lb.

Work Done by Gravity

The work done by or against gravity is then merely the weight of the body handled, in pounds, multiplied by the *vertical* distance moved, in feet.

So when water falls in any manner from one level to another, gravity does work on the water equal to the product of the weight of water falling by the difference between the original and final levels. If no attempt is made to recover this work, it is all eventually turned into heat by friction and eddy currents.

This heating effect is not appreciable because a comparatively large amount of work corresponds to a small (as we judge it) amount of heat. If no heat were lost by radiation or evaporation and all the heat generated by the fall remained in the water, a stream of water would have to fall 778 ft. to be raised one degree in temperature.

Difference Between Power and Work

It will be noted that the discussion so far has been mainly confined to *work*, whereas *power* is usually the thing of most interest in connection with a hydraulic plant. Power relates work and time. It is the rate of doing work and may be expressed directly in foot-pounds per second. If 1,000,000 lb. of water flow over a falls in 5 seconds and the available fall is 100 ft., the *work* available in this period is 100,000,000 ft.-lb., but the available *power* is 100,000,000 ÷ 5 =20,000,000 ft.-lb. per second. Horsepower is merely another way of stating the same thing, one horsepower being 550 ft.-lb. per second. In the case just mentioned the horsepower of the *falls* would be 20,000,000 ÷ 550 = 36,363.

Cubic Feet per Second

The flow of water in water-power installations is almost universally referred to as a certain number of cubic feet per second rather than pounds per second. Since one cubic foot equals 62.4 lb. of water, the theoretical power produced by one cubic foot per second falling 10 ft. would be 62.4 × 10 ÷ 550 =1.134 hp. If this is multiplied by 0.88, the product is unity. That is equivalent to saying that if a plant has an over-all efficiency of 88 per cent, the power practically available amounts to one horsepower for each cubic foot per second falling 10 ft. Thus an eighty-foot fall of 9,000 cu-ft. per second would develop 9,000 × 8 = 72,000 hp. at this efficiency, which is not far from that of most large plants. If the actual efficiency is known, the power will be the product of 0.1134, the discharge in cubic feet per second, the head in feet and the efficiency. Thus 5,000 cu-ft. per second falling 60 ft. and used with an efficiency of 80 per cent would develop 0.1134 × 5,000 × 60 × 0.8 = 27,216 hp.

The Essentials of Efficiency

It is, of course, the aim of the station designer to secure as high an over-all efficiency as is consistent with reasonable cost. All the power "in" the water is delivered to the feeders except that which disappears as heat. So though the hydraulic designer may never think of heat as such, his real method of producing high efficiency is to prevent, wherever possible, the generation of heat. If this is accomplished, high efficiency automatically follows.

The main causes of heat production—and hence the points that must be attacked to secure good efficiency—are bearing friction, electrical resistance in generator, skin friction of water on all wetted surfaces where there is rapid relative movement, internal friction within the water due to swirls and eddies, and velocity remaining in water discharged from the draft tube.

Which Losses Are Important?

Of these the first two are relatively unimportant. There is but little room for improvement in the matter of bearing losses and electrical losses. Skin friction is of considerable importance since any hydraulic installation has large areas of metal in contact with moving water. This loss increases rapidly with the velocity, so its reduction is largely a matter of keeping the velocity low wherever practicable. This in turn is dependent upon providing large areas for the flow of water. In the turbine itself the flow area is necessarily somewhat restricted so that here the possibility of reducing skin friction must be balanced against the other factors, the final result being a compromise, as is the case with most engineering designs.

The most difficult thing in hydraulic design is to cut down the loss caused by the internal friction between particles of water in the swirls and eddies. Swirls and eddies are reduced to a minimum by making the passages that conduct the water smooth and uniform and by avoiding sudden changes in diameter or direction.

Function of the Draft Tube In Utilizing Velocity and Head

The discharge from the turbine itself is usually at a fairly high velocity and at a point above the level of the tail water. To make full use of the available head, a draft tube is required. This serves two purposes. In the first place it utilizes the fall (if any) from the turbine discharge to the tail-water level. Its second purpose is to remove most of the velocity from the water in such a way that the corresponding energy is returned to the turbine rather than dissipated as heat by fluid friction and skin friction. This second purpose is served by expanding the draft tube in such a way that it slows up the water continuously and without shock. The proper shape for this tube has been, and is still, a subject of much controversy among hydraulic engineers but all agree that the object to be attained is the slowing up of the water with the minimum of disturbance.

As a matter of fact great progress has been made until today the efficiencies of the newer hydraulic power plants are so high as to leave little room for future improvement.

New and Improved Equipment

Electrically Operated G. E. Flow Meter

A new type of electrically operated flow meter has recently been developed that operates on the principle of an alternating-current transformer. The meter can be made indicating, recording or integrating to measure the flow of gas or liquid through a pipe. Owing to the electrical principle of operation the instruments can be placed any distance away from the pipe line the flow through which is being metered. Fig. 1 shows a schematic diagram of the metering equipment. The electrical element is a transformer. On one leg of a laminated-iron core is the primary winding, which is connected to a source of alternating current of constant potential. Without any secondary load on the transformer, the only current flowing in the primary winding is that to magnetize the core and supply the losses in the core and coil. If a properly calibrated ammeter is connected in the circuit as shown, it will indicate this current. On the other leg of the core is a fiber container made in two compartments—one, *A*, that surrounds the core and another, *B*, that contains mercury but is in communication with compartment *A*. These compartments are connected on the opposite sides of an orifice in the pipe line through which the fluid or gas flows. With no flow in the pipe line the pressure in *A* and *B* will be the same, and the mercury is at a height so as to be at a level with the bottom of *A*. When there is a flow in the pipe line, a difference in pressure is created across the orifice, so that the pressure in *A* will be lower than in *B*, resulting in mercury being caused to rise up into *A* to a height that will equalize the pressure between the two chambers. When the mercury rises in chamber *A*, it surrounds the laminated core and forms a short-circuited secondary winding of one turn. This ring of mercury around the core is similar to a copper ring, except that it has a higher resistance for a given cross-section. In the mercury around the core a current will be induced just as in a closed secondary winding on a transformer. To supply the current in the secondary, an increase of current must flow in the primary, which will be indicated on the ammeter. The difference between the two currents will be a measure of the fluid flowing in the pipe line, since the height of the mercury in chamber *A* is a measure of the quantity. The current flowing in the mercury ring will be in direct proportion to its height. That is, a ring one-quarter inch deep will have one-half the current flowing in it that a ring one-half inch deep would have, since the former has twice the resistance of the latter. Increasing the current in the secondary would cause proportional increase in the primary current. The ammeter could be made to indicate the quantity of fluid directly by adjusting it so that with no flow in the pipe line its needle would point to zero. Then calibrate it so that, as the mercury rises in chamber *A*, the increased current flow in the primary will cause the needle to point to divisions on the scale representing the flow in the pipe.

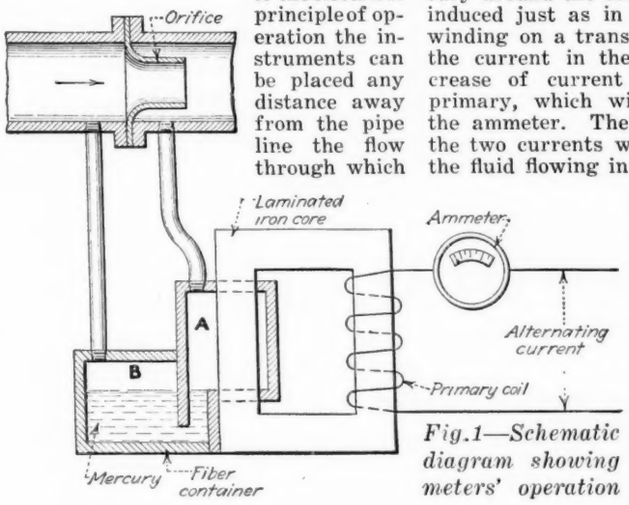


Fig. 1—Schematic diagram showing meters' operation

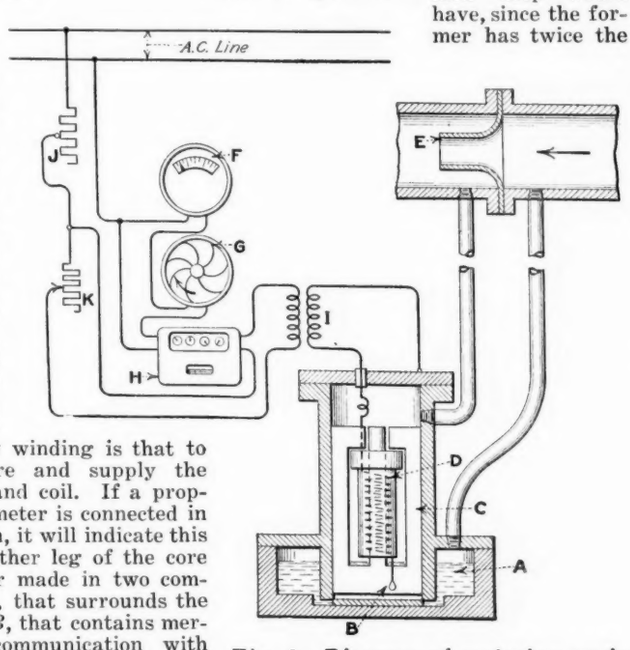


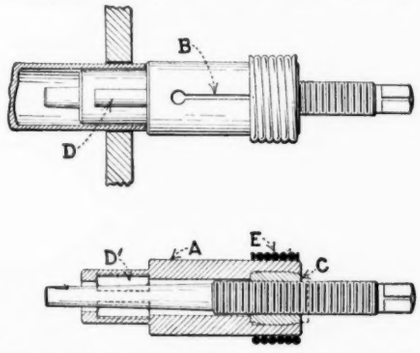
Fig. 2—Diagram of metering equipment and electrical connections

A, mercury; *B*, ground connection of primary coil; *C*, transformer core; *D*, primary coil; *E*, flow nozzle producing a differential pressure; *F*, indicating ammeter; *G*, curve-drawing ammeter; *H*, watt-hour meter; *I*, 108- to 5-volt transformer; *J*, voltage regulator resistance in series with line; *K*, adjustable resistance compensating for voltage drop in line.

A diagram of the commercial form of meter is given in Fig. 2. In this arrangement the fiber container is replaced by a cast-iron meter body which contains a three-legged core-type transformer *C*, with the primary winding *D* on the middle leg and the mercury *A* contained in the bottom of the meter body. A flow in the pipe line will cause the mercury to rise into the middle chamber to form a secondary winding around the primary coil and cause a current to flow in the meters in proportion to the flow in the pipe line. A step-down transformer *I* is connected between the meters and the meter body to insulate the latter from the former and to supply a current at less than 5 volts to the primary coil in the meter body. A potential of 108 volts is applied to the primary of the transformer and is held constant by a voltage regulator. Meters *F* and *G* are indicating and curve-drawing ammeters which are calibrated to indicate and record the instantaneous quantities of fluid or gas flowing through nozzle *E*. The volume flowing for a given time is recorded on the watt-hour meter *K*. This new meter is the product of the General Electric Co. and is not intended to supersede the mechanically operated flow meter of this company.

Record Tube Expander

The illustration shows a type of tube expander that has been used extensively in Europe and is now being introduced in this country by the Techno-Service Corp., New York City.



Expanded mandrel is fed automatically

This expander differs from the usual type in that an automatic feed is

provided for the mandrel. This feed also serves as a means of indicating when the expanding process is complete and tends to prevent any undue straining of material in the tube.

Referring to the illustration, the expanded body *A* has three slots in the outer end as at *B* and is recessed to take the nut *C*, which is slightly tapered at both ends. After this nut is in place, the clamping spring *E* is put on. The expander is fitted with three parallel rollers in the usual way.

In the process of expanding a tube, a certain amount of pressure is exerted on the nut *C*, which, being transmitted to the cylinder, is limited by the spring collar *E*. When the joint between the tube and plate is tight, an excess pressure is set up and the feeding of the mandrel is retarded, which action causes the nut *C*, to move outward. The rolling is then continued until the nut projects about $\frac{1}{8}$ in., as shown by dotted lines in the lower figure, at which point the rolling process is complete. The expander is released from the tube by screwing back the mandrel and permitting the nut *C* to return into the cylinder.

The expander is particularly adapted for being driven by an electric or air drilling machine, in which case the power is transmitted to the mandrel by means of a driving shaft fitted with a flexible joint.

Stanwood Down-Draft Firebox Boiler

A type of boiler designed for power or heating services in which a down-draft furnace has been combined with a horizontal tubular shell, has recently been developed by the Houston, Stanwood & Gamble Co., Cincinnati, Ohio.

The boiler is a plain cylindrical horizontal fire-tube with only one pass for the gases from the furnace to the smokebox. The tubes are centrally located, giving liberal steam space at the top and water space at the bottom, the tubes constituting almost the entire heating surface of the boiler. A manhole is provided in the rear head below the tubes and another in the top of the shell near the front head.

The furnace is directly in front of the boiler and is of the down-draft type. It consists of the usual primary upper grate of water-tube grate bars and a secondary shaking grate. The water-tube grates lead directly from a horizontal manifold at the front of the furnace, into the front head of the boiler above the tubes. Large feed or circulating pipes lead from the bottom of the manifold to the front boiler head below the tubes.

The furnace is located in a steel extension, riveted or bolted to the boiler shell, and is lined with firebrick and

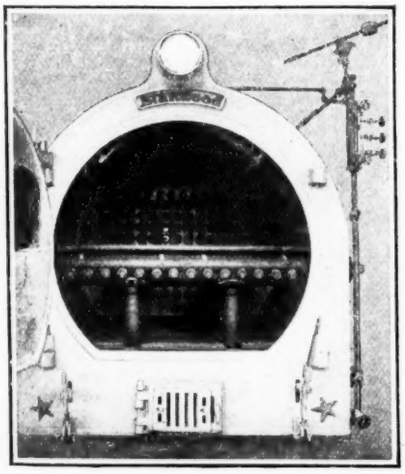


Fig. 1—Front view of boiler

insulating brick on each side, and with firebrick only on the inside of the furnace front. The upper fuel chamber above the water-tube grates has an arched top. A portion of this arch is made of firebrick. The top or arch keys are made of cast iron, each one is hollow and has a large number of openings through which the air for combustion is admitted. This distribution of air tends to keep the arches cool and the air is also preheated before it reaches the fuel bed. The air is admitted, as indicated by the arrows, through a centrally located intake on the top of the furnace controlled by a damper.

The employment of water-tube grates tends to produce a rapid circulation within the boiler; the water in the lower part of the shell passes through the feed pipes and is induced in an upward current into the manifold by the heat absorbed by the water in the water-tube grates, from which it is delivered into the boiler near the top surface of the tubes.

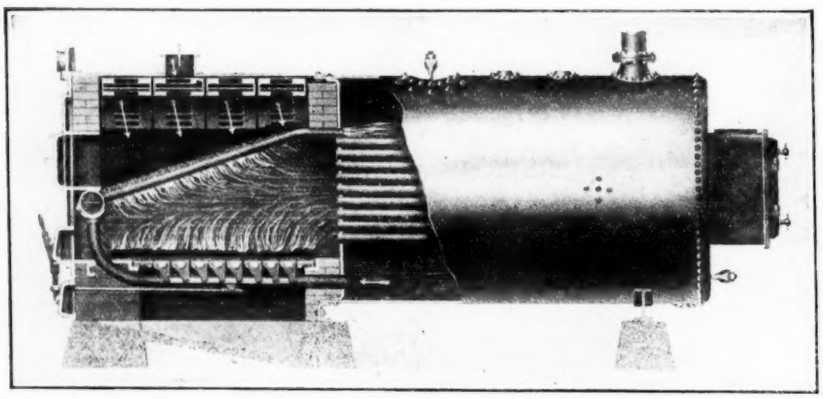


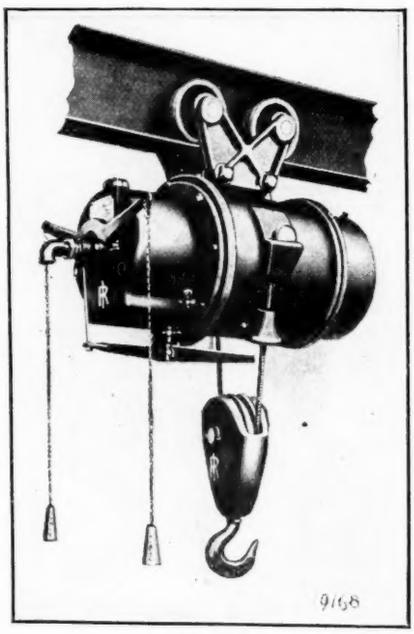
Fig. 2—Shows arrangement of water-tube and shaking grates

It will be seen that there are no riveted seams exposed to the fire, and with the exception of the heads there are no flat surfaces to be stayed. Handholes are provided at each end of the manifold and brass plugs in its shell opposite each water-grate tube for inspection. These boilers are regularly built in sizes from 300 to 2,000 sq.ft. of heating surface for working pressures of 15 to 125 lb. or constructed to order for pressures up to 200 lb.

New Line of Air Motor Hoists

A line of new air motor hoists, including five different sizes with capacities ranging from 500 to 10,000 lb., is a recent addition to air-operated equipment manufactured by the Ingersoll-Rand Co., New York.

A balanced three-cylinder motor is used, which operates in either direc-



A motor hoist equipped with monorail trolley

tion and which can be throttled down slowly at all loads. The motor is geared through a mechanical train to a hoisting drum. A safety stop lever is provided, which closes the throttle and stops the motor whenever the load is by chance raised to the top of the lift.

The hoists are equipped with an automatic brake, which functions and holds the load at any desired position whenever the hoist is not operating or whenever the air supply is cut off by throttling or otherwise. The brake is automatic in action and requires no attention from the operator.

The hoist is equipped with a roller-bearing monorail trolley or top hook mounting as desired. As shown in the illustration, the trolley mounting reduces considerably the headroom necessary for installing. The motor and gears are entirely inclosed, making the hoist suitable for use in all kinds of weather or in a moisture- or acid-laden atmosphere.

Piping Specifications

By H. W. Brooks

PIPING specifications published to the industry for the most part have been those of the Power Piping Manufacturers' Association or of individual manufacturers of piping. To a large extent they represent the manufacturer's viewpoint. As an example of specifications prepared from the purchaser's viewpoint, the accompanying specifications on the high-pressure steam header for the Batavia power house of the Chicago, Aurora & Elgin Railroad Co. (described in *Power*, Oct. 30 and Dec. 18, 1923) are presented in the hope that they may be of some service in a general way for similar work. They were prepared to withstand a total temperature of 650 to 700 deg. F. in case additional superheaters are later installed, although the initial operating temperature is only 567 deg. F.

HIGH-PRESSURE STEAM HEADER

Part I—General

A. Working Conditions—All material shall be suitable for steam at 200 lb. per sq.in. gage pressure and 175 deg. F. superheat.

B. Facing and Drilling (For flanges on pipe, valves or castings)—Dimensions shall be arranged in all cases by the contractor where necessary so as to match up exactly with the parts of the existing high-pressure header and piping which will be retained in service. The contractor shall send his engineer or inspector to the Batavia power house to check all such dimensions and shall be responsible for the proper fitting and operation of the new fittings in connection with the old. In all other cases where it is not necessary that fittings or parts match up with existing fittings, they shall be according to the latest American standard for 250 lb. pressure. All flanges shall be extra-heavy, long-hub, cast steel. All drilled holes shall be located accurately, correct in size, perfectly cylindrical, and at right angles to the flange face. No plugged holes will be allowed. The backs of all flanges shall be spot-finished around the bolt holes to such a depth that the metal left is $\frac{1}{16}$ in. less in thickness than the flange itself. In all cases the required finish shall include machining the face of the flange, drilling and spot facing the bolt holes as specified previously and finishing the gasket surface as specified in the next paragraph. All flange faces shall be protected properly from rust and damage in transit.

C. Gasket Surface (All flanges except loose flanges)—The gasket surface shall form an angle of 90 deg. with the pipe axis. The finishing cut shall give a spiral or concentric ring effect, the pitch of which shall not exceed $\frac{1}{31}$ in. radially and the depth not be over $\frac{1}{63}$ in. No defect of any kind shall exist on the gasket surface on final inspection. The contractor shall paint all gasket surfaces at his plant with one coat of white lead and oil for protection from the elements.

D. Visual Inspection — All pipe, flanges, valves, castings, etc., shall be arranged as to size and shape so as to match up perfectly in every case with parts of the existing header, which will be retained. In all other cases dimensions shall be arranged according to the latest American standard for 250 lb. pressure or according to the dimensions for corresponding parts shown in the latest catalog of the Crane Company. In the absence of a distinct specification for material of a different grade, size or shape, all material furnished shall conform to the standard specifications for power piping adopted by the Power Piping Society on June 24, 1915, and revised June 27, 1922. The contractor will be responsible for the correctness, matching up and fitting of all dimensions, both on the parts of the present header to be retained, and valve dimensions (in case contract for valves is let separately from contract for piping and fittings) so as to produce a complete successfully operating header with a minimum of joints, or it is to be understood distinctly that the contractor's inspector or engineer is to be sent to Batavia to check up all dimensions and that the purchaser will not take any responsibility for the correctness of dimensions given or as shown on blueprints supplied, both of which shall be understood to be approximate only. All machined parts shall be accurate in dimensions with a tolerance in no case to exceed $\frac{1}{16}$ in. All materials must be sound, reasonably smooth and free from cracks and injurious defects.

E. Acceptance and Tests—No materials will be accepted which develop flaws, cracks, etc., during the process of bolting up or when steam at working pressure is turned on. Failure to meet this requirement shall be deemed sufficient cause for rejection.

Part II—Pipe and Vanstoning

A. Pipe

Manufacture—(1) All pipe for vanstoning up to and including 12-in. pipe size shall be selected full-weight lap-welded steel pipe. Fourteen-inch size and larger shall be selected outside diameter (O.D.) lap-welded steel pipe of the following thickness, within a tolerance of $\frac{1}{64}$ in. plus or minus.

14-in. O. D. pipe	$\frac{3}{8}$ in.
16-in. O. D. pipe	$\frac{7}{16}$ in.
18-in. O. D. pipe	$\frac{1}{2}$ in.
20-in. O. D. pipe	$\frac{11}{16}$ in.

(2) All pipe shall be rolled from steel made by the open-hearth process and shall conform to the National Tube Company's specification for "line" pipe.

Length—(3) Each pipe bend or straight piece shall be made up from a single random pipe length finished to the maximum length which the parts of the present header will allow. No intermediate flanges or weld shall be allowed throughout the length unless such are specifically agreed to by the purchaser. In case the contractor is unable to supply the required length of pipe in a single piece, he shall so advise the purchaser within one week after receipt of order.

B. Vanstoning

Upsetting and Flaring—(1) The vanstoning shall be done at a temperature between a light yellow (1,079 deg. C. or 1,975 deg. F.) and a full red (746 deg. C. or 1,375 deg. F). In the process of vanstoning, the pipe ends shall be upset sufficiently to obtain a thickness in the turned-over portion which after machining will not be less than the wall thickness of the untreated pipe. The vanstoning shall be done in a manner such that the machined gasket surface will intersect the bore of the pipe at right angles and make what is termed a "square" Van Stone joint. On completion of the vanstoning the pipe end shall be annealed by heating it to a temperature expressed in terms of color as a salmon (899 deg. C. or 1,650 deg. F.).

All Van Stone joints shall be full Van Stone extending to the inside of the bolt hole.

Machining (2)—The turned-over flange shall be machined on the gasket surface as specified in Part I (C). The back of this flange shall be smooth and true with the loose flange face, except for a fillet where it joins the pipe wall. All Van Stone joints will be machined both back and front in order to give proper bearing surface for the flange.

(3) If the vanstoning process materially decreases the inside diameter of the pipe, the contractor shall bore it out to the diameter of the untreated pipe. The wall thickness after boring shall not be less than that of the untreated pipe.

Welding—(4) At the discretion of the purchaser pipe which split in the turned-over end during vanstoning, may be repaired by welding. After welding, the entire end of the pipe shall be annealed as specified in Part II, B (1).

C. General Requirements

(1) All requirements laid down in Part I of these specifications shall be complied with.

Part III—Forged Steel Flanges

A. Manufacture

Ingots—(1) Ingots may be made by the open hearth, electric furnace, or any other method approved by the purchaser.

(2) Sufficient discard shall be made from each ingot from which the forging slab or bar has been procured, to secure freedom from piping and undue segregation.

Forging—(3) All forging shall be done at temperatures between a light yellow (1,079 deg. C. or 1,975 deg. F.) and a full red (746 deg. C. or 1,375 deg. F.). The dies must be kept free from scale, and no work must be done on metal to which any considerable quantity of scale is attached.

Annealing—(4) Forgings may be annealed by heating to a salmon color (899 deg. C. or 1,650 deg. F.) at the discretion of the contractor.

Machining—(5) The flanges shall be machined as specified in Part I. All unfinished surfaces shall be reasonably smooth.

B. Chemical Properties and Tests

Composition—(1) The phosphorous content shall not exceed 0.05 per cent and the sulphur content shall not exceed 0.05 per cent. The general com-

position of the metal shall be characteristic of the best high-grade forging steel.

C. Physical Properties and Tests

Tension Tests—(1) The forgings shall conform to the following minimum requirements as to tensile properties:

Tensile strength, lb. per sq.in.	60,000
Yield point (tensile) lb. per sq.in.	0.45
Elongation in 2 in. per cent	20
Reduction in area (desired 40 per cent) per cent	30

(2) The yield point shall be determined by the drop of the beam on the testing machine as specified by the A.S.T.M.

Bend Test (3)—Bend-test specimens shall bend cold (without hammering) through 120 deg. around a 1-in. pin, without cracking on the outside of the bent portion.

Test Specimens (4)—The above tests may be made at the discretion of the purchaser and at his expense. Test specimens, procured from such portions of flanges as the purchaser desires, shall be machined to the dimensions established by the A.S.T.M. Tension-test specimens shall be shaped to fit the holders of the testing machine in such a way that the load will be axial. Bend-test specimens shall be machined to 1 x ½ in. as specified, with corners rounded to a radius not over $\frac{1}{16}$ in.

D. General Requirements

All forgings shall conform to the requirements laid down in Part I of these specifications.

Part IV—Cast-Steel Fittings and Fitting Flanges

A. Manufacture

Process—(1) The steel may be made by the electric furnace, open hearth, or converter process.

Annealing—(2) All castings shall be annealed by heating to a temperature as close as possible to 875 deg. C. (1,507 deg. F.). Castings shall be kept at the maximum temperature a sufficient length of time to insure refining the grain throughout the entire cross-section of the heaviest segment. Following this heating the castings shall be cooled slowly and uniformly in the furnace.

Sand Blast—(3) All castings shall be sandblasted or otherwise suitably cleaned after annealing.

Machining—(4) Machining shall conform in all respects with the requirements specified in Part I. Special castings shall be machined as specified in Part I.

Boring—(5) All fittings shall be bored out concentric with the bolt circle to a diameter $\frac{1}{64}$ in. of the port opening specified and to a depth of not less than 6 in. from the flange face (except in the case of an increasing bore) or where the castings are less than 10-in. size. Where the bore reduces, the counterbore shall follow the same taper. Where the bore increases, the counterbore shall be straight and shall extend in until the tool is clear. When the boring is finished, the walls of the fitting shall not vary more than $\frac{3}{4}$ in. from specified thickness.

Unfinished Surfaces—(6) The unfinished surface in the bore of all fittings shall be cast against a smooth core with a view to leaving as frictionless a passage for the steam as possible.

Painting—(7) The contractor shall not paint any surfaces of his work except the gasket faces. These he shall paint at his plant with one coat of white lead and oil for protection from the elements, or such other protective coating as may meet the approval of the purchaser. All shop paint shall be thoroughly dry before the material painted leaves the contractor's plant.

B. Chemical Properties and Tests

Required Composition—(1) The phosphorous content as determined above shall not exceed 0.06 per cent and the sulphur content shall not exceed 0.07 per cent. The general composition of the metal shall be characteristic of the best high-grade cast steel.

C. Physical Properties and Tests

Tension Tests—(1) The castings shall conform to the following minimum requirements as to tensile properties:

Tensile strength, lb. per sq.in.	60,000
Yield point (tensile), lb. per sq.in.	0.45
Elongation in 2 in., per cent	20
Reduction in area (desired 40 per cent) per cent	30

(2) The yield point shall be determined by the drop of the beam of the testing machine as specified by the A.S.T.M.

Bend Test—(3) Bend-test specimens shall bend cold (without hammering) through 120 deg. around a 1-in. pin without cracking on the outside of the bent portion.

Test Specimens—(4) The above physical tests may be made at the discretion of the purchaser and at his expense on tension- and bend-test specimens obtained from test bars cast as described in Part III, Section C (4). Tension- and bend-test specimens shall be machined to the dimensions established by the A.S.T.M. Tension-test specimens shall be shaped to fit the holders of the testing machine in such a way that the load will be axial. Bend-test specimens shall be machined to 1 x ½ in. as specified, with corners rounded to a radius not over $\frac{1}{16}$ in.

Percussion Test—(5) A percussion test shall be made on each casting. During this test each fitting shall be suspended by chains and hammered all over with a hammer whose head weighs not less than five pounds. If in any casting, cracks, flaws, defects or weakness appear after such treatment, the casting shall be rejected.

D. Additional Tests

Hydrostatic—(1) All fittings shall be tested to and shall sustain without leakage a hydrostatic test of 800 lb. per sq.in.

Steam—(2) All castings shall be given a steam test conforming to the conditions of service; that is, steam at a gage pressure of 200 lb. per sq.in. superheated 175 deg. F. This test, if so desired, will be made by the purchaser after receipt of castings and at no expense to the contractor.

Rejection—(3) Any defects developing as a result of the tests described in Section D (1 and 2), may be treated by suitably applied welding, if such meets with the approval of the purchaser. The purchaser shall determine whether or not castings so treated shall be re-annealed. If so, the method described in Section A (2) shall be employed. All castings developing defects under these tests, and not restored, shall be rejected. The report of such rejection shall be made within one week after the failure.

F. General Requirements

(1) All castings shall conform to the requirements laid down in Part I of these specifications.

Welding Defects—(2) No castings shall be filled or welded in any way until after inspection by the purchaser. Nor shall castings be offered for inspection if covered with paint, rust or any other substance to such an extent as to hide defects.

(3) Minor defects which do not impair the strength of the casting, may, with the approval of the purchaser's inspector, be welded by an approved process. The defects shall first be cleaned out to solid metal, and after welding, the casting shall be reannealed as specified in Section A (2), if the purchaser so desires.

Part V—Semi-Finished Machine Bolts

Spcial Steel

Physical Properties—(1)

	Minimum
Elastic limit, lb. per sq.in.	55,000
Tensile strength, lb. per sq.in.	90,000
Elongation in 2 in., per cent	20
Reduction in area, per cent	40
Brinell hardness number	200 to 350

The bolt shall bend cold through 120 deg. F. about an inner diameter equal to that of the bolt.

Chemical Properties—(2) Neither the sulphur content nor the phosphorous content shall exceed 0.04 per cent. The general composition of the metal shall be characteristic of the best grade special bolt steel. Heat treatment to be such as will produce the above-specified physical properties.

Finish—(3) The head and nut faces shall be true for wrench fit within $\frac{1}{32}$ in. from U. S. standard bolt dimensions. The underside of heads and nuts shall have a smooth tool finish.

Note: All bolts are subject to inspection and test.

Failure to meet any one of the requirements specified may be deemed sufficient cause for rejection.

Part VI—Special Flange Steel, Monel Mounted, Gate Valves, Suitable for a Working Steam Pressure of 200 lb. Gage and 175 deg. F. Superheat

Material	Steel body, Monel mounted
Style	O. S. Y. or postyoke gate
Body	Cast Steel
Mounting	Monel, renewable
Stem	Steel
Flanges	Faced and drilled either to match existing parts or according to American standard for 250 lb. with $\frac{1}{16}$-in. raised face, inside bolt holes, in case it is unnecessary to match existing parts.
Bypass	8-in. sizes and larger to be fitted with bypass and tap for ¾-in. drain.

Part VII—Stop and Check Valves

Special flange steel, monel mounted, stop and check valves suitable for a steam working pressure of 200 lb. gage and 175 deg. F. superheat.

Style	Flange body swing check
Body	Steel
Mounting	Monel seat and disk-renewable
Flanges	Faced and drilled and arranged to fit existing parts or else arranged according to latest American standard for 250 lb. with $\frac{1}{16}$ in. raised face, inside bolt holes, in case it is unnecessary to match existing parts.

German Engineers Discuss High-Pressure Steam

THE general interest in higher steam pressures was again shown in the high-pressure steam convention which the V.D.I. (Institute of German Engineers) held in Berlin Jan. 18 and 19, approximately two thousand engineers from all parts of Germany and foreign countries attending.

The president of the V.D.I., Prof. G. Klingenberg, in the opening speech, said that by utilizing the most recent improvements in design and operation, we are in a position to raise the economy of steam plants practically to the same level as that of Diesel engines and to reduce the coal consumption to about one-half of the present figures.

Director Münzinger, of Berlin, then discussed the "technical and economical principles of the production and utilization of high-pressure steam." He said that for a station, whose steam is used exclusively to produce power, the economically most favorable result is obtained when the pressure is raised to approximately 500 to 710 lb. Beyond this pressure the additional costs due to the heavier boiler drums exceed the possible saving. The lecturer therefore advocated boiler designs with water and steam chambers limited as far as is compatible with reliability and the production of dry steam.

Effect of Back Pressure

If a power plant supplies large quantities of steam against high back pressure for manufacturing purposes, as is often the case for instance in the chemical industry, a saving of 10 to 20 per cent on the running expenses can be made, even when the boiler pressure has been increased to 1,400 lb. This shows that high-pressure service will for the time being be most advantageously applied to combined heating and power plants operating with back pressure. In purely power-generating stations, however, a saving of approximately 25 per cent of the thermal value of the fuel consumed can be made with a slight increase in pressure, by adopting reheating and the stage heating of feed water by bled steam.

The next lectures given by Professor Goerens, of Essen, and Director Loch, of Düsseldorf, dealt with the question of materials and the manufacture of steam boilers for high pressures. Professor Goerens chiefly pointed out the advantages accruing from the employment of nickel steel, the greater mechanical strength of this material permitting the boiler drums to be made of thinner plates, which are more easily fabricated. In addition he held that the nickel steels with a 3 and 5 per cent content of nickel are preferable to the mild steel prescribed by laws, because when the boiler is subjected to higher temperatures, they do not show such a reduction of the mechanical strength or such increased brittleness as mild steel. Professor Goerens recommended making high-pressure boiler drums of one single forged or rolled piece, which can be subjected to the necessary hydraulic tests before delivery, to ascertain if it has the requisite mechanical strength.

The second day's meeting was likewise opened by a short speech of the president, who suggested that the High-Pressure Steam Convention accept a pressure of 500 lb. as a temporary limit and establish standards for boiler dimensions on this basis.

Safe Construction of Boilers

Dr. M. Guilleaume, of Merseburg, reported interesting and important experience gained in steam-boiler plants, citing tests made by the Association of Large Boiler Owners, founded in 1920 to promote the safety of large boiler plants. These tests proved that the utmost care must be exercised when testing the boiler plates, so that the designer may be convinced that the plates in their entirety possess all the qualities that were ascertained during the tests, and, furthermore, that the working of the plates until they are assembled in the finished boiler, especially too high a riveting pressure, can materially affect the properties of the plates and render them unreliable.

He said that the observations made with special measuring devices of novel design, to ascertain the alterations of shape to which the finished boiler is subjected owing to fluctuations of the temperature and pressure, are of great value. These alterations can never be entirely avoided in actual service owing to the fluctuations of the load, therefore it is essential to design the boilers so that they have sufficient elasticity and that no undue stresses arise in any of the parts. On the other hand, one should also strive to eliminate such influences by keeping the load on the boilers and feed-water supply as constant as possible. Valuable observations have also been made concerning the circulation of the water in boilers. The results of these observations are of special interest now that it is planned to run boilers at much higher pressures than has been the custom.

The last speaker, Prof. E. Josse, of Charlottenburg, lectured on the properties and utilization of high-pressure and superpressure steam. He told the audience that when the working pressure is raised from 280 to 1,400 lb., the most recent data concerning the properties of steam proved that theoretically the useful work available in a given weight of steam could be increased by 15 to 83 per cent according to the back pressure carried. However, if this is to be applied to a steam engine or steam turbine, two points must be observed. In the first place high-pressure steam becomes saturated much more quickly when expanding, because considerations of safety limit the maximum temperature. In the second place the work to be produced by the steam in the high-pressure zone will be greater the higher the initial pressure. The first point makes it compulsory to dry the steam repeatedly during the different stages of expansion up to back pressure, as otherwise the water contained in the steam would reduce the efficiency with which the turbine can use the available heat drop. The second point, he said, is of special importance for steam turbines, the present design of which (contrary to that of reciprocating engines) is not suited to utilize the steam economically in the high-pressure zone. As a matter of fact, recent high-pressure steam technique has been the incentive to develop special types of turbines which economically utilize high-pressure steam and are the first practical results of modern high-pressure practice. A number of works are already busy building such turbines, and the extensive tests made with a high-pressure turbine designed by the Erste Brünner Maschinenfabriks-Aktiengesellschaft, showed an efficiency of over 80 per cent, an extraordinary good performance.

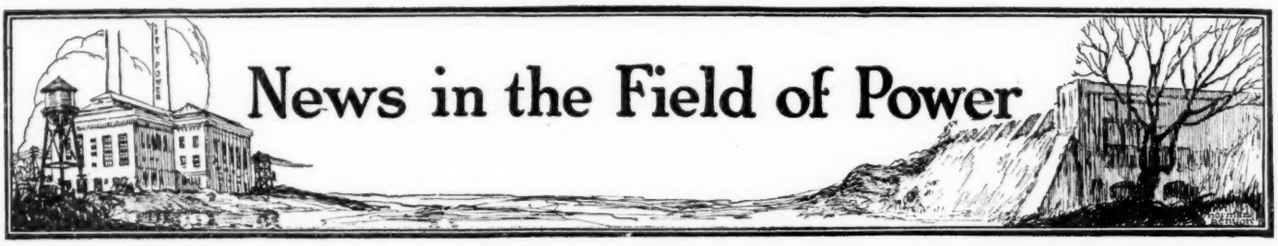
Heat Transmission at Penn. State

At the recent annual meeting of the American Society of Heating and Ventilating Engineers, in New York, a paper was presented by P. Nicholls, describing his work with the heat meter in the Research Laboratory of the American Society of Heating and Ventilating Engineers, at Pittsburgh. An abstract of this paper appeared in the Feb. 5 issue.

Among the discussions following this paper was one covering the heat-transmission work done during the past few years at the Pennsylvania State College. This discussion was omitted from the report of the meeting, for lack of space, so a few of its more important points will be presented here.

The original heat-transmission laboratory was built in 1910, the first experiments being made with calorimeters of the box type. A report on this method showed that the box method was open to severe criticism for general use. When this became evident, plans were made by the engineering experiment station for the construction of a guarded hot-plate, 2 ft. square. This gave improved results. A still further advance was made by the adoption of heat meters, similar to those described by Mr. Nicholls. The hot-plate is now being used for standardizing or calibrating these meters.

The calibration of four ½-in. cork heat meters is virtually completed. Check tests made months apart have given consistent results, showing the permanency of the plate. The calibration of four additional plates, two of ebony-asbestos wood and two of transite, is now under way. The application of heat meters for laboratory use is also being tested in connection with materials of known conductivity against a tank containing crushed ice. The investigations have been going forward steadily, although many interruptions have interfered with the calibration of these plates. Present progress of the work indicates that the results will be available within a few months.



President Appoints St. Lawrence River Commission of the United States

Hoover Heads This Commission Also the American Part of the Joint Engineering Board

THE Canadian Government, in a note made public by the State Department on March 14, agrees to the suggestion of the United States government that two additional engineers be appointed by each government to the Joint Engineering Board which will have charge of the consideration of the St. Lawrence River waterway project.

At the same time President Coolidge announced that he has appointed the St. Lawrence Commission of the United States, comprising: Herbert Hoover, Secretary of Commerce, chairman; William C. Breed of New York, former president of the New York Merchants' Association; James E. Davidson, of Bay City, Mich.; James P. Goodrich, of Winchester, Ind., former Governor of Indiana; James R. Howard of Chicago, former president of the Farm Labor Bureau Federation; James D. Noonan, of the American Federation of Labor; Stephen B. Davis, of Washington, counsel; Charles P. Craig, of Duluth, Executive Secretary; an additional member will be chosen from New England.

In commenting on the appointment of the St. Lawrence Commission, Mr. Hoover explained that the commission's work will be confined entirely to a study as to the proper time to undertake the development of the stream. There no longer is question, he said, as to the advisability of developing both the waterway and the power resources of the river. The only question is when the work should begin. He anticipates some continuance of the opposition from New York and Buffalo to the waterway feature, but he declared that any losses these cities might suffer by reason of the waterway would be compensated by the advantages that would come to them by the development of the river's power possibilities.

Mr. Hoover laid emphasis on the fact that there is at this moment a market for the 2,000,000 hp. which can be developed in connection with the project. He pointed out that this fact must be taken into consideration in comparing this project with most large hydroelectric developments where large blocks of power necessarily must be brought in prior to the development of a market to absorb the energy.

It is Mr. Hoover's idea that the commission should conduct hearings to ascertain whether the work should begin in the near future, or whether plans should be made with the idea of beginning the work ten or twenty years hence. Before the public hearings are begun, the commission will meet probably about April 1 to formulate the plans best calculated to attain the objectives it is seeking.

Secretary Hoover emphasized the point that this commission is dealing with American phases of the question only, and its work must not be confused with the engineering board, of which he also is head. The Army engineers who are to work under his direction will co-operate with Canadian engineers to revise the estimates of the cost of the project. This step is being taken at the request of the Canadian government. The cost contained in the exhaustive reports which have been made on this project are based on prices in effect at the times the reports were made. It is believed that substantial changes in the figures will result from their revision. Secretary Hoover made it particularly clear that there is to be no new engineering investigation of the project. Three exhaustive reports have been made by as capable men as could be secured to make a new report. The work which Mr. Hoover will direct will be confined entirely to a revision of cost figures.

New Generating Station for New York Edison Co.

The New York Edison Co., a subsidiary of the Consolidated Gas Co., has plans for the erection of the largest generating station in the country which will have an ultimate capacity of 500,000 kw. This station will be located at Fourteenth St. and the East River. The power house will embody in its design all the latest and best practices. It is expected that it will be ready for operation in 1926.

New York Protests Lakes Drainage

C. S. Ferris, representing the attorney-general and the Water Power Commission of New York, recently notified the Committee on Rivers and Harbors that the state authorities are opposed to the Hull bill giving legal sanction to the diversion of water from Lake Michigan for purposes of draining and sanitation for Chicago.

Mr. Ferris said at a public hearing that he would join with the states of Michigan, Minnesota, Indiana and Ohio in making a protest against such diversion of waters on the ground that it menaced navigation interests by lowering lake levels. He said that New York was opposed to the bill on the further ground that it curtails power developments in the Niagara River. Chicago representatives deny both contentions.

It has been said that the Chicago Sanitary District is now diverting water from Lake Michigan up to a maximum of 10,000 ft. per sec. instead of 4,617 ft. per sec. as authorized in a permit issued by the Secretary of War. The city was recently enjoined from the use of the higher volume of water and an appeal to the Supreme Court is pending.

Interconnection Effected Between Five States

Ten independent electric light and power companies, serving Ohio, Pennsylvania, Maryland, Virginia and West Virginia, are bringing into being, through the completion of a series of co-operative agreements for the mutual exchange of power, a superpower system which may be described as the coal-field group, covering the area between Cleveland, Erie, the coal fields along the Kentucky-West Virginia state line south of Charleston, W. Va., and a point just outside of Baltimore, and including the entire Pittsburgh district. The participating companies have a combined capitalization of $315,000,000 and a total generating capacity of more than 2,000,000 hp. produced largely by plants in the coal regions.

A.I.E.E. Has Two Nominees for President

At the meeting of the Board of Directors of the American Institute of Electrical Engineers held in New York March 14, the report of the Committee of Tellers of its canvass of the nomination ballots cast for candidates for the offices falling vacant July 31 was read.

There were two candidates for the presidency: Charles E. Skinner, of Pittsburgh, who received 2,448 votes, and Farley Osgood, of Newark, N. J., who received 2,238 votes. The Board decided not to select a directors' nominee for the office of president, and the names of both elegible candidates will be placed on the final ballot.

The board selected the following list of directors' nominees for the other offices to be filled: For vice-presidents: Harold B. Smith, Worcester, Mass; L. F. Morehouse, New York City; Edward Bennett, Madison, Wis.; H. W. Eales, St. Louis, Mo.; John Harisberger, Seattle, Wash. For managers: John B. Whitehead, Baltimore, Md.; E. B. Merriam, Schenectady, N. Y.; J. M. Bryant, Austin, Tex. For treasurer, George A. Hamilton, Elizabeth, N. J.

Interconnection Studies and Maps Progressing

Large Power Plants Will Be Located as Near Coal Mines as Condensing Water Supply Will Permit

SPECIAL studies being made of the superpower situation at the instance of Secretary Hoover, are revealing that the progress in interconnection of electric lines is proceeding at an amazing rate. The extent to which interconnection has been taking place has not been appreciated entirely because this is the first time that data covering this situation throughout a large area, have been brought together at one place. The information submitted by the states has been combined with that of the federal government and mapped. These maps, which are being prepared by the Federal Power Commission, are destined, it is believed, to be the focusing point of a large amount of attention soon.

The rapid trend toward interconnection has an important bearing on the coal industry. Its chief importance unquestionably is that it is making for consolidations and concentrations within the industry which is moving more and more rapidly toward a vast public service. The importance in that connection is the ruling of the Supreme Court of the United States in the Kansas industrial court case. In that decision the court implied that the only kinds of business in which the state may step in and compel arbitration, the acceptance of awards and other drastic forms of regulation, are public utilities.

It already is indicated, according to the engineers who are studying this problem, that some day in the future the great service of supplying power will be located largely in great plants as near the source of coal supply as a supply of condensing water will permit. The great agencies that will distribute this power are not going to invest millions of dollars in steam plants without insuring themselves absolutely as to coal supply. The steel companies have long followed the policy of purchasing reserves of coke and coal. It is certain that these great central stations are going to safeguard their supplies of steam coal in much the same fashion. In that connection it may be remarked that the study of interconnection has progressed far enough to indicate that chief reliance in the industrial area of the Atlantic Seaboard must be placed on electricity generated at steam plants.

The language of the decision can be construed to mean that the state court may still hold that the Kansas industrial court has the power to jail a man for inciting a strike even though it has not the power to compel arbitration. Whether the state court will insist on its former view, in spite of the Supreme Court's decision, remains to be seen. So far as the layman can penetrate the mysteries of the legal mind, it seems apparent that the Supreme Court still believes that the production of bituminous coal is not a public business and that the public cannot interfere to fix the price of the commodity or to compel men to cease striking. Whatever the court may have said about the relatively unimportant mines of Kansas, probably would not be repeated were it dealing with a great service of coal and power such as those toward which we are headed in the North and East.

Feeder Short-Circuit Ties Up Chicago Stations

Soon after 10 p.m., on March 17, a short-circuit in one of the main feeders from Calumet station in Chicago, Ill., put that plant out of service. As a safety measure the company also shut down the Fisk St. and Quarrie St. plants on either side of the Chicago River. That left only the Northwest plant, in Roscoe St. and a few minor generating stations to carry the load for the entire city.

The greater part of the south side of the city was thrown into darkness and other southern districts were affected. Theater and motion-picture audiences had to find their way to the street through unlighted aisles, and entertainments and dances as well as private homes, had to resort to gas and candles. The city's pumping plants at Twenty-second St., Sixty-ninth St. and Chicago Ave. were put out of commission.

The company's direct-current area, including the "Loop" was unaffected. After two hours of work part of the Calumet plant was restored to service.

Increase in Municipal Public Utility Plants in U. S. Shown

The Department of Commerce presents preliminary figures from the forthcoming quinquennial census report on central electric light and power stations for the United States. These figures cover both commercial and municipal plants, both those generating current and those distributing purchased current, but do not include those operated by mines, factories, hotels, and other enterprises not in the nature of public utilities, which consume the current generated, nor plants operated by the Federal Government and State institutions. The "number of establishments" refers to ownership or control and in many cases an "establishment" represents more than one generating station.

The following table shows the data for 1922 in comparison with 1917:

	1922	1917	Per Cent of Increase (1)
Number of establishments (2)	6,356	6,542	− 2.8
Commercial	3,775	4,224	−10.6
Municipal	2,581	2,318	11.3
Total horsepower of prime movers	20,296,235	12,936,755	56.9
Steam engines and turbines:			
Number	6,525	7,487	−12.8
Horsepower	14,171,222	8,449,076	67.7
Internal-combustion engines:			
Number	3,256	2,934	11.0
Horsepower	302,995	210,406	44.0
Waterwheels and turbines:			
Number	3,481	3,374	3.2
Horsepower	5,822,018	4,277,273	36.1

(1) A minus sign (−) denotes decrease.
(2) The decrease in the number of establishments is due chiefly to consolidations and the extension of transmission systems.

Engineers' Report on Colorado Basin Reaches Congress

DECLARING that the time had come for the determination of a definite government policy covering all phases of the development of the Colorado River Basin, the Secretary of the Interior on March 17 sent a report to the chairman of the Senate and House Committees on Irrigation and Reclamation of Arid Lands covering the Swing-Johnson bill now before Congress.

Accompanying it were the findings and conclusions of a select board recently appointed and composed of engineers Spencer Crosby, Corps of Engineers, U. S. Army; W. E. Weymouth, E. B. Debler, Walter R. Young, of the Reclamation Bureau; Herman Stabler, the Geological Survey, and W. Kelly, of the Federal Power Commission, with detailed technical observations.

In a discussion of the proposed Boulder Dam, power plant with transmission lines, and the All-American Canal for supplying the Imperial Valley with irrigation water, it was asserted that this project was of such magnitude as to challenge the country's ablest engineers and should be the subject of most deliberate review by Congress. Attention was called to the fact that the Boulder Dam as outlined will be the largest engineering structure of its kind ever attempted, containing 3,375,000 cu.yd. of concrete, or three times as much as the Assuan Dam in Egypt, which is the greatest masonry dam ever built; that it proposed to raise the water surface 605 ft., a height greater than the Washington Monument, and create a reservoir 120 miles long, storing one and one-half times as much water as that of the Gatun Lake on the Panama Canal.

The total cost of the three features of the dam, transmission lines and All-American Canal, should be estimated at $200,000,000, which includes a margin of safety to protect the government.

As to the magnitude of the proposition, it was said that whether or not it is practical at any cost to divert water through tunnels in the canyon walls of the river, whether a mass of masonry unapproached in size in the history of engineering is practical and whether it is possible to give more than an intelligent guess of its cost are problems not to be passed upon by one man alone. It was urged that Congress in its final decision upon a plan of development for the Colorado River, consider all the separate and distinct features including flood control, impounding water for farming, storing water for generating electric power, the All-American Canal and the possible future necessity for domestic water to supply

California cities. It was recommended that such commensurate measures should be initiated by Congress from a point of view of 50 and 100 years hence rather than now.

Foreign Service Bill Meets Opposition

The Committee on Interstate and Foreign Commerce of the House of Representatives is disposed to press for an early vote on the bill which gives a permanent legislative status to the foreign service of the Department of Commerce. Unexpected opposition arose to the measure from other departments of the government based on the fear that the Department of Commerce might encroach upon some of the prerogatives of the foreign service of other departments.

The feeling among the members of the House committee, however, is that the better service which this will give American business and American interest generally far outweighs the trepidation of other departments as to a possible technical invasion of their respective jurisdictions abroad. Apparently the bill does not meet the approval of the American Protective Tariff League, as it has given publicity to adverse comment on the measure.

Connecticut "Get Together" a Success

The Garde Hotel in New Haven was the rendezvous for Connecticut Members of the A.S.M.E. on the evening of March 18. With the New Haven Section as hosts and a good representation from other centers in the state, the annual "get together" was highly successful not only as a social gathering with good food and entertainment, but also in its more serious aspects.

Guided by H. R. Westcott as toastmaster, the engineers listened to an inspirational speech by John L. Davis, of New Britain, on "America the Land of Opportunity." Following this Dr. Ira M. Hollis, past-president of the A.S.M.E., spoke on "The Part That Production Plays in the Permanency of Nations." Dr. Hollis delved into history to show that engineering skill and the resulting productiveness were the essential factors in the permanency of nations. In Egypt and Rome, for example, the nation flourished in proportion to its production and began to die when engineers sank into the background. Production has three divisions, manufacturing, agriculture, and distribution, in each of which the engineer must continue to play a leading part if the country is to progress. Dr. Hollis expressed the belief that the future of New England industry lies mainly in the intensified production of small finished articles of commerce. Development of power resources is of prime importance in this connection. The weak spot of the engineer, said Dr. Hollis, is his individualism. He urged as an essential to the welfare of this country the more active co-operation of engineers in guiding affairs of state and suggested methods of getting technical students to think along this line.

Coming Conventions

American Association of Oil Burner Manufacturers. No. 518 Bank of Galesburg Bldg., Galesburg, Ill. First annual meeting at Hotel Chase, St. Louis, Mo., April 1-3.

American Chemical Society. Charles L. Parsons, 1709 G St. N.W., Washington, D. C. Annual meeting at Washington, D. C., April 21-25.

American Electrochemical Society. C. G. Fink, Columbia University, New York City. Meeting at Hotel Bellevue-Stratford, Philadelphia, Pa., April 24-26.

American Institute of Electrical Engineers. F. L. Hutchinson, 29 West 39th St., New York City. Spring convention at Birmingham, Ala., April 7-10.

American Society of Civil Engineers. John H. Dunlap, 29 West 39th St., New York City. Spring meeting at Atlanta, Ga., April 9-12.

American Society of Mechanical Engineers, Calvin W. Rice, 29 West 39th St., New York City. Spring meeting at Cleveland, Ohio, May 26-29.

American Water Works Association. J. M. Diven, 153 West 71st St., New York City. Annual convention at New York City, May 19-21.

American Welding Society, Miss M. M. Kelly, 29 West 39th St., New York City. Annual meeting at New York City, April 23-25.

Association of Iron and Steel Electrical Engineers. W. M. Chandler, 708 Empire Building, Pittsburgh, Pa. Fuel Saving Conference—Combustion Engineers, April 2-3 at Wm. Penn Hotel, Pittsburgh, Pa.

Eastern Ice Manufacturers' Association. W. H. Ross, 35 Warren St., New York City. Spring meeting at Hotel Adelphia, Philadelphia, Pa., April 2-3.

Electric Power Club. S. N. Clarkson, Keith Bldg., Cleveland, Ohio Meeting at Seaview Golf Club, Absecon, N. J., May 26-29.

International Railway Fuel Association. J. B. Hutchinson, 6000 Michigan Ave., Chicago. Annual convention at Hotel Sherman, Chicago, Ill., May 26-29.

Master Boiler Makers Association. H. D. Vought, 26 Cortlandt St., New York City. Meeting at Chicago, May 20-23, 1924.

National Association of Stationary Engineers. Fred W. Raven, 417 South Dearborn St., Chicago, Ill. Annual convention and exhibition at Hotel Pantlind, Grand Rapids, Mich., Sept. 8-13. Annual conventions and exhibitions of the state associations are scheduled as follows: **Kansas Association** at Parsons, Kans., April 16-18. J. M. VanSant, 739 Horne St., Topeka, Kan. **Indiana Association** at Lafayette, Ind., May 5-6, Prof. A. W. Cole, Purdue University, Lafayette, Ind. **New Jersey Association** at Newark, June 6-8. Joseph P. Flynn, 612 Franklin St., Elizabeth, N. J. **Iowa Association** at Sioux City, June 10-14. Abner Davis, 16 Waterhouse Block, Cedar Rapids, Iowa. **Ohio Association** at Akron, June 19-21. T. S. Garrett, 2622 East Second St., Dayton, Ohio.

National Electric Light Association. M. H. Aylesworth, 29 West 39th St. New York City. Annual convention at Atlantic City, Young's Million-Dollar Pier, May 19-24. **Nebraska Section,** H. M. Davis, Banker's Life Bldg., Lincoln, Neb. Sectional meeting at Omaha, May 8-9. **Southwestern Division,** S. J. Ballinger, San Antonio Public Service Co., San Antonio, Texas. Divisional meeting at New Orleans, April 22-25.

Society of Industrial Engineers. George C. Dent, 608 South Dearborn St., Chicago, Ill. Eleventh annual convention at Hotel Statler, Buffalo, N. Y., April 30 to May 2, 1924.

World Power Conference. O. C. Merrill, Federal Power Commission, Washington, D. C. British Empire Exhibition, Wembley, London, June 30-July 12.

Society Affairs

The Fort Wayne, Ind., Section of the A.I.E.E. will on April 17 hear R. F. Schuchadt speak on "Public Utilities."

The Hartford Engineers' Club and the **A. I. E. E.** will hold a meeting on March 27, at which Mr. Ferguson will speak on "St. Lawrence River Power."

The Ontario Section of the A.S.M.E. will hear H. M. Horsfall speak on "The Manufacture and Uses of Steel Wire and Cable" at its April 2 meeting.

The Metropolitan Section of the A.S.M.E. will have as the topic of its April 1 meeting "Modern Tendencies in Marine Propulsion," by Emil Berg, of the General Electric Co.

The Syracuse Section of the A.S.M.E. will have as the topic for its meeting on March 27, "Leather Belting." Louis Arny, of the Leather Belt Exchange, will be the speaker.

The Los Angeles Section of the A.S.M.E. will hear, on the evening of March 27, at Bull Pen Inn, H. L. Doolittle speak on the "Design of Penstocks for Hydro-Electric Plants."

The Advisory Committee of the National Exposition of Power and Mechanical Engineering has been enlarged and now consists of: Fred R. Low, James Partington, Nevin E. Funk and Calvin W. Rice of the A.S.M.E.; Walter H. Johnson of the N.E.L.A.; Fred Felderman of the N.A.S.E.; Homer Addams of the A.S.H.V.E.; George A. Horne of the A.S.R.E.

Business Notes

The Bury Compressor Co., Erie, Pa., announces the new location of its New York office at 50 Church St., which will be in charge of H. L. Dean.

The McClave-Brooks Co., Scranton, Pa., announces that Arthur S. Thomas assumed the duties of manager of its New York City offices, 461 Eighth Ave., on March 1.

The Brown Instrument Co., Philadelphia, Pa., announces that T. R. Harrison, formerly associated with the pyrometry department of the Bureau of Standards, has assumed charge of the research department of the Brown Instrument Co.

The Yarnall-Waring Co., manufacturers of power plant devices, Chestnut Hill, Philadelphia, Pa., announces that Harold S. Webster has joined its organization as district manager for eastern Pennsylvania, southern New Jersey, Delaware, Maryland, District of Columbia and Virginia. He succeeds George W. Hill, Jr.

The Raymond Brothers Engineering Co., Chicago, designer of "Rayco" pulverized fuel systems, etc., announces the removal of its general sales office to the Combustion Engineering Corp. Bldg., 43 Broad St., New York City. H. D. Savage, vice-president of the Combustion Engineering Corp. will be in charge of sales.

Trade Catalogs

Motors—The Triumph Electric Co., Cincinnati, Ohio. Leaflet describing four types of small motors.

Recorder, Helical Tube—The Foxboro Co., Inc., Foxboro, Mass. Leaflet "The Improved Helical Tube Movement," describes with illustrations this recorder.

Washers, Air—Clarage Fan Co., Kalamazoo, Mich. Catalog No. 71 describes with illustrations and tables of general data these type "V" air washers.

Power Plant, General Purpose—The Smith Gas Engineering Co., Dayton, Ohio. Bulletin No. 20 describes two small power plants for use where 17 to 25 hp. requirements are desired.

Insulation, Heat—The Philip Carey Co., Lockland, Ohio. Bulletin No. 102 "Heat Losses from Bare Iron Surfaces," contains tables and curves showing losses. Bulletin No. 103 "Heat Transmission Tests," contains data on commercial heat insulating materials with curves showing the effect of thickness of covering, etc.

Fuel Prices

BITUMINOUS COAL

The following table shows the trend of the spot steam market in various coals (mine run bases, f.o.b. mines):

Coal	Market Quoting	Mar. 10, 1924	Mar. 17, 1924
Pool 1	New York	$3.25	$3.25
Smokeless	Columbus	2.25	2.25
Clearfield	Boston	2.40	2.60
Somerset	Boston	2.60	4.75
Kanawha	Columbus	1.65	1.65
Hocking	Columbus	2.00	2.00
Pittsburgh No. 8	Cleveland	1.90	1.90
Franklin, Ill.	Chicago	2.50	2.50
Central, Ill.	Chicago	2.25	2.25
Ind. 4th Vein	Chicago	2.50	2.50
West Ky.	Louisville	1.90	1.90
S. E. Ky.	Louisville	2.00	2.00
Big Seam	Birmingham	1.85	1.85

FUEL OIL

New York—March 20, light oil, tank-car lots, 28@34 deg. Baumé, 5c. per gal., 36@40 deg. 5½c. per gal., f.o.b Bayonne, N. J.

St. Louis—March 11, tank-car lots, f.o.b. St. Louis; 24@26 deg., $1.90 per bbl.; 26@28 deg., $2 per bbl.; 28@30 deg., $2.05 per bbl.; 32@36 deg., gas oil, 5½c. per gal.; 36@40 deg., 6½c. per gal.

Pittsburgh—March 8, f.o.b. local refinery, 30@34 deg., fuel oil, 5½c. per gal.; 36@40 deg., fuel oil, 6c. per gal.

Cincinnati—March 18, tank-car lots, f.o.b. local refinery, 24@26 deg. Baumé, 6c. per gal.; 26@30 deg., 6⅛c. per gal.; 30@32 deg., 6¼c. per gal.

Dallas—March 14, f.o.b. local refinery, 26@30 deg., $130 per bbl.

Philadelphia—March 14, 28@30 deg., $2.31@$2.36 per bbl.; 18@22 deg., $2.10 @$2.16½; 13@16 deg., $1.68@$1.74½ per bbl.

Boston—March 18, tank-car lots, f.o.b. heavy oil, 12@14 deg. Baumé, 4⅝c. per gal., light oil, 28@32 deg. Baumé, 6⅝c. per gal.

New Plant Construction

Calif., Fort Miley — The quartermaster, Fort Mason, plans for an early call for bids for a 250 gal. per min. motor-driven centrifugal pump and accessory equipment, for the local water station.

Calif., Smartsville — The Yuba River Power Co., will begin work early in April on its proposed dam on the Narrows of the Yuba River, near Smartsville, for hydro-electric power development, proceeding with the initial unit of the power station at a later date. The project will involve about $5,000,000. Lars Jorgenson, Hobart Building, San Francisco, is engineer.

Fla., Miami — The director of public Utilities, Ernest Cotton, is reported to be considering the installation of a water-softening plant for municipal waterworks service.

Fla., Tampa — The Florida-West Indies Corp., recently formed with a capital of $3,000,000, plans for the construction of a power house at its proposed furniture manufacturing plant in this section, to cost close to $500,000. W. F. Miller is president.

Ga., Coreen — The Clinchfield Portland Cement Co., Kingsport, Tenn., is perfecting plans for a power house at its proposed local plant, on which work will be commenced at an early date. It will cost in excess of $900,000, with equipment.

Ga., Thomaston—The Flint River Power Co., an interest of the Central Georgia Power Co., Macon, Ga., has preliminary plans under way for the construction of a hydro-electric generating plant in this section, where site has been selected. It will cost in excess of $500,000, with equipment. The parent company is said to be planning for extensions in its central station at Macon.

Idaho, Weippe—The Coeur d'Alene Mill Co. plans for the construction of a boiler plant at its proposed local lumber mill, estimated to cost about $65,000. Fred Herrick is general manager.

Ind., Indianapolis—The Board of Public Works has plans under way for additions and improvements in the municipal market building, to include the installation of a new ice and refrigerating plant and system. Frank B. Hunter, State Life Building, is city architect.

Ind., Indianapolis — The Board of Trustees, City Hospital, is planning for extensions and improvements in the boiler plant at the institution, to include the installation of two additional boilers and accessory apparatus, water-softening equipment, etc.

Ind., Indianapolis — The Pitman-Moore Co., 111-17 North Capitol Avenue, manufacturer of serums, etc., plans for the installation of a refrigerating plant at its new three-story works to be constructed at Morris Street and Madison Avenue, estimated to cost $250,000. James E. Bartlett is president.

Iowa, Dubuque—The Board of Works is said to be planning for the installation of electric-operated pumping machinery in connection with proposed extensions and improvements in the waterworks, for which a bond issue of $125,000, has been approved by the City Council.

Iowa, Eagle Grove—The Common Council has approved the immediate construction, by day labor, of a new one-story pumping plant, 26 x 37 ft., for the municipal waterworks. Electric-operated equipment will be installed. J. K. Baker is city clerk.

Iowa, Greenfield—The City Council has approved an ordinance for a bond issue of $45,000, to be submitted to voters at a regular city election, the proceeds to be used for extensions in the municipal electric plant and installation of additional equipment. H. G. Spooner is city clerk.

Iowa, Mapleton—The Iowa Light, Heat & Power Co., Carroll, has acquired the local power plant, heretofore owned and operated by Max McGraw. The new owner will consolidate with its other stations, and plans for expansion in this section.

Kan., Newton—The City Council is planning an early call for bids for the construction of its proposed one-story steam-operated power plant for municipal service. E. T. Archer & Co., New England Building, Kansas City, Mo., are engineers.

Kan., Reserve—The Common Council, G. A. Hall, city clerk, is considering the installation of electric-operated pumping machinery at the proposed municipal waterworks. The Shockley Engineering Co., Graphic Arts Building, Kansas City, Mo., is engineer.

Ky., Whitesburg—The Coeburn Lumber Co. plans for the construction of a boiler plant at its proposed local mill, to cost close to $50,000. L. F. Jackson is company engineer.

La., Loreauville—The Loreauville Sugar Co., Inc. is planning for the purchase of a 60 kw., 3-phase, 60-cycle, 440-volt generator, with exciter, switchboard and instruments, and auxiliary apparatus, for installation at its power house. A number of motors, 5, 20 and 40 hp. will also be purchased.

La., Monroe—The Missouri-Pacific Railroad Co., St. Louis, Mo., has tentative plans for the rebuilding of the power plant and portion of shops at its local repair works, recently destroyed by fire with loss estimated at $100,000. E. A. Hadley is chief engineer.

La., New Orleans—The Standard Rubber Co., J. H. Cooper, president, is considering the construction of a boiler plant at its proposed local mill for the manufacture of automobile tires, etc., to cost approximately $100,000.

La., Shreveport—The Arsenic Compound Co. plans for the construction of a boiler plant at its proposed local factory, estimated to cost in excess of $65,000. T. B. Weatherby is president; and H. S. Lonergan, treasurer.

Md., Hagerstown—The Tidewater Steel Co., Baltimore, M. D. Perine, president, recently organized, plans for the construction of a power house at its proposed local rolling mill, for which a site has just been acquired. It is estimated to cost close to $200,000.

Mexico, Puerto—The Ford Motor Co., Highland Park, Detroit, Mich., plans for the construction of a power house at its proposed local assembling plant, estimated to cost $450,000.

Mich., Menominee—The Northern Paper Mills Co., Green Bay, Wis., has acquired a local water power site from the White Rapids Paper Co., Oshkosh, Wis., and has preliminary plans under way for the construction of a hydro-electric generating plant.

Minn., Fergus Falls—The City Council has approved a measure for a bond issue of $600,000, to be submitted to voters at an election, April 1, the proceeds to be used for a municipal power plant. Plans will be prepared at an early date. H. J. Collins is city clerk.

Minn., Faribault—The Common Council has tentative plans under advisement for the construction of a municipal electric power plant, and expects to arrange a bond issue at an early date.

Minn., Little Falls—The Common Council has authorized the purchase of a site for the erection of a new pumping station for the municipal water system, for which a list of pumping machinery and auxiliary equipment will be arranged at an early date. Andrew Johnson is city clerk.

Minn., Marietta—The Common Council plans to call for bids about April 1, for pumping machinery and auxiliary equipment for the proposed municipal waterworks. W. E. Buell & Co., Davidson Building, Sioux City, Iowa, are engineers.

Minn., Owatonna — A. L. Mullergren, Kansas City, Mo., engineer for the proposed municipal power plant, is arranging a list of equipment, to include steam turbo-generators or uniflow engines (still to be decided); two 300 hp. watertube boilers; open feed water heater; spray nozzle cooling system; boiler feed pumps and auxiliary apparatus, with 150 ft. high concrete stack, and water-treating plant. The plant will cost $350,000. C. J. Servatius is city clerk.

Miss., Natchez—The Natchez Ice Co. has started work on a new addition to its ice and refrigerating plant, to double, approximately, the present output of 40 tons per day. Electric equipment, with Diesel engine, will also be installed for local service.

Mont., Miles City—The Chicago, Milwaukee & St. Paul Railroad Co., 80 West Jackson Boulevard, Chicago, Ill., has tentative plans under consideration for the construction of a local electric power plant, estimated to cost $60,000.

Mont., Ryegate — The Golden Valley County Board of Commissioners plans for the installation of electric-operated pumping equipment in connection with its proposed Franklin irrigation project, on which work will be commenced during the spring. W. O. Wood is county clerk.

Neb., Crab Orchard—The Common Council has called a special election on March 25 to vote bonds for $14,500, for extensions and improvements in the municipal electric power plant.

N. J., Bayonne—The Asiatic Petroleum Co., Bergen Point, has filed plans for the construction of a one-story boiler plant at its local works, estimated to cost $25,000. The Moody Engineering Co., 90 West Street, New York, is engineer.

N. J., Burlington—The City Council will call for bids early in April for a new pumping plant for sewerage service, in connection with additions to the system to cost $30,000. Sherman & Sleeper, Fifth and Cooper Streets, Camden, N. J., are engineers.

N. J., Morris Plains—The Commissioner of Institutions and Agencies, State Office Building, Trenton, has extended the date for receiving bids for the proposed 1,700 hp. boiler plant at the local state hospital from March 26 to April 2. Plans and specifications at the office of the Division of Architecture and Construction of the Department.

N. J., Whippany—The Agar Mfg. Corp., 167 Forty-first Street, Brooklyn, N. Y., manufacturer of corrugated paper products, plans for the construction of a boiler plant at its proposed local mill on site recently acquired, to cost close to $100,000, with equipment.

N. Y., Brooklyn—G. W. Ethbert, 355 East Sixteenth Street, has had plans completed and will soon commence work on a one-story cold storage plant at 746-62 Forty-second Street. J. W. Magrath, 367 Fulton Street, is architect in charge.

N. Y., Oswego—The Standard Oil Co. of New York, 26 Broadway, New York, is said to be planning the installation of electric-operated pumping equipment at its proposed oil storage and distributing plant on local site, and similar plant at Sacketts Harbor, N. Y., to cost in excess of $1,200,000.

N. Y., Syracuse—The Syracuse Lighting Co., South Warren Street, formed by a recent merger of the company of the same name and the Onondaga Utilities Co., is reported to have plans under consideration for the construction of a steam-operated electric power plant on local site.

N. C., Asheville—The Uneka Mining Co., recently organized with a capital of $50,000, is planning to purchase an air compressor and other power equipment for installation at its local properties. James E. Rector, 11 Church Street, attorney, represents the company.

N. C., Greensboro—The Rucker Bonded Warehouse Co. is considering the erection of a cold storage and ice plant on local site, to cost about $75,000. J. M. Workman is company engineer.

N. C., Hendersonville—The Bacon Hosiery Mills are planning for the installation of oil burners and oil-burning equipment at their boiler plant, replacing present coal operation.

N. C., North Wilkesboro—E. P. Blair, North Wilkesboro, is forming a company to construct and operate a local ice-manufacturing plant to cost about $40,000. S. V. Tomlinson is also interested in the organization.

N. C., Rougemont—W. A. Carver, Rougemont, is planning for the purchase of a water wheel and other equipment for a local power installation.

N. C., Salisbury—The Citizens' Ice & Coal Co. is reported to be planning for extensions in its ice and refrigerating plant, including the installation of additional equipment. The company has recently increased its capital from $75,000 to $450,000, for expansion.

N. C., Sanatorium—The Board of Trustees, North Carolina Sanatorium, Dr. T. W. M. Long, chairman, will receive bids until April 1, for equipment for extensions to the waterworks at the institution, including electric-operated centrifugal pumps and accessory equipment; steel water tank and tower. The Gilbert C. White Co., Durham, N. C., is consulting engineer.

N. D., Ellendale—The City Council will receive bids until April 7, for one electric-operated centrifugal pump, with capacity of 700 gal. per min., pumping against a head of 150 ft.; 50 hp. motor, direct-connected. Edward A. Smith is city auditor, in charge.

Ohio, Canton—The Wilson Rubber Co. is reported to be planning for the construction of a boiler plant at its proposed four-story mill, estimated to cost $70,000. Work will begin at an early date.

Ohio, Dayton—The Green & Green Co. will build a power house at its proposed three-unit baking plant, estimated to cost $450,000. Work in the initial unit, comprising a five-story structure, 150 x 160 ft., will soon be commenced. Schenck & Williams, Dayton, are architects. Weston Green is president.

Ohio, Lancaster—The Hocking Glass Co. is considering the construction of a boiler plant in connection with the proposed rebuilding of the portion of its local works destroyed by fire, March 7, with loss in excess of $500,000, including equipment. I. J. Collins is president.

Ohio, Toledo—The H. J. Heinz Co., Progress Street, Northside, Pittsburgh, Pa., plans for the construction of a power house at its proposed local food products plant, on site selected at Oak Street and Whitmore Road, to cost in excess of $750,000.

Okla., Bartlesville—The City Council is arranging a special election to vote bonds for about $1,000,000 for waterworks extensions, to include the construction of a dam on Butler Creek, 50 ft. high and 1,000 ft. long, and the installation of an electric-operated pumping station with rated capacity of 5 million gal. per min. The Burns & McDonnell Engineering Co., Interstate Building, Kansas City, Mo., is consulting engineer.

Okla., Cushing—The City Council is considering the installation of electric-operated pumping machinery in connection with extensions and improvements in the waterworks and sewerage plant, estimated to cost $500,000.

Ore., Medford — The Talent Irrigation District is considering the installation of electric-operated pumping equipment in connection with an irrigation project covering 4,500 acres, including the construction of a dam on Emigrant Creek. Bonds for $440,000 have been voted.

Ore., Terrebonne—H. V. Gates, and associates, have mad application to the state water department for permission to utilize waters from Crooked Creek for a hydro-electric development, estimated to cost $100,000.

Pa., Connellsville—The Baltimore & Ohio Railroad Co., Baltimore, Md., plans for the construction of a water treatment plant for locomotive boiler service near its local shops, taking supply from the Youghiogheny River, with installation to include 3 electric-operated pumping units with capacity of 500 gal. per min., and auxiliary apparatus. The estimated cost is placed at close to $100,000.

Pa., Philadelphia — The depot quartermaster, United States Marine Corps, will take bids until March 28, for the conversion of existing boilers for oil feed and operation; also, for a concrete fuel oil storage tank, with alternate bid on steel tank.

Pa., Tobyhanna — The quartermaster, Tobyhanna artillery range, will take bids until April 1, for two centrifugal pumps for installation in existing power house; elevated steel water tank, and auxiliary equipment.

Pa., Weatherly—The Pennsylvania Power & Light Co., Allentown, Pa., is opening negotiations for the purchase of the local municipal electric plant, and plans for extensions in operations in this vicinity. Superstructure work is under way on its new hydro-electric generating plant at Hawley, near Honesdale, Pa., to cost in excess of $500,000.

R. I., Newport—The United States Navy Supply Office will take bids at once for 700 lb. flexible metallic packing, as specified in Ord. Req. 202.

S. C., Charleston—The Gulf Refining Co., Frick Annex, Pittsburgh, Pa., plans for the construction of a boiler plant and pumping station at its proposed oil storage and distributing plant in the Shipyard Creek section, estimated to cost $800,000, with machinery. W. B. Lamb is superintendent of construction.

Tenn., Cookeville—The Common Council is considering the installation of electric-operated pumping machinery in connection with extensions in waterworks and sewerage plant, for which a bond issue of $100,000, has been sold.

Tenn., Nashville—The Nashville Pulp & Paper Co., plans for the erection of a power house at its proposed local pulp mill, estimated to cost $1,000,000.

Tex., Dallas—The Board of Works has preliminary plans under advisement for the installation of a hydro-electric power station in connection with the new city reservoir, with initial capacity of 2,500 hp. The complete project will cost close to $5,000,000.

Tex., Lone Oak—The Common Council is considering the installation of electric-operated pumping machinery in connection with waterworks extensions to cost $50,000, in which amount bonds have been sold.

Tex., Pearsall—The Texas Central Power Co., San Antonio, plans for extensions in the local power plant, recently acquired, including the installation of a 160 hp. Diesel engine and auxiliary equipment.

Va., Chatham—The City Council will receive bids until March 31, for equipment for a municipal waterworks, including low service motor-driven centrifugal pump, direct-connected; low service centrifugal pump, belt driven; two high service centrifugal pumps, similar drives; semi-Diesel fuel oil engine, with air starter outfit; 150,000 gal. capacity steel tank on tower, and all accessory apparatus. William M. Tredway is city clerk; the Ambler Engineering Co., Richmond, Va., is engineer.

Va., Reusens—The Lynchburg Traction & Light Co., Lynchburg, Va., has completed plans for additions to its local power plant to increase the capacity to 8,000 hp. Contracts have been placed for a portion of the equipment and other apparatus will soon be contracted for.

Wash., Hillyard—The Sinclair Refining Co., 111 West Washington Street, Chicago, Ill., plans for the installation of electric-operated pumping equipment at its new local storage and distributing plant, estimated to cost $150,000.

Wash., Kalama—J. G. Cruver, chairman of the port district, has made application to the state water department for permission to use waters from he Kalama River, for a hydro-electric development, totaling 23,660 hp., estimated to cost $550,000.

Wash., Longview—Charles C. Garland, Tacoma, Wash., engineer, has plans in progress for the construction of a dam and hydro-electric power plant on the Toutle River, near Castle Rock, to cost close to $5,000,000, with equipment. Bids will be asked in the near future.

Wash., Seattle—The Log Cabin Products Co., St. Paul, Minn., plans for the construction of a power house at its proposed local branch plant for the manufacture of maple sugar products, on site now being selected, estimated to cost $300,000. Frank I. Towle is president.

W. Va., Buckhannon—The Monongahela-West Penn Public Service Co., West Penn Building, Pittsburgh, Pa., has acquired the power plant and property of the Buckhannon Light & Power Co. Plans are under consideration for extensions in the station, including the installation of additional equipment.

W. Va., Oak Hill—W. C. Hayes has tentative plans under way for the erection of a one-story ice-manufacturing plant on local site. G. C. Seibt, Rose Turner Building, Beckley, W. Va., is engineer.

W. Va., Wheeling—The Kablitzer Packing Co., 4128 Water Street, plans for the erection of a six-story addition, 100 x 100 ft., to its ice and cold storage plant, estimated to cost $150,000 with equipment. It is expected to commence work early in May. George N. Kablitzer is president.

Wis., Athens—The Village Council is considering the installation of electric-operated pumping machinery at the proposed waterworks and sewerage plants additions, estimated to cost $60,000. W. G. Kirchoffer, 22 East Carroll Street, Madison, Wis., is engineer.

Wis., Rhinelander—The Common Council is reported to be considering the installation of electric-operated pumping equipment in connection with proposed waterworks extensions and improvements. Anna Moe, city clerk.